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JUN 8 1935

Heating · Piping and Air Conditioning

Pulling up flanged joint by heating bolts with acetylene torch—see page 303.

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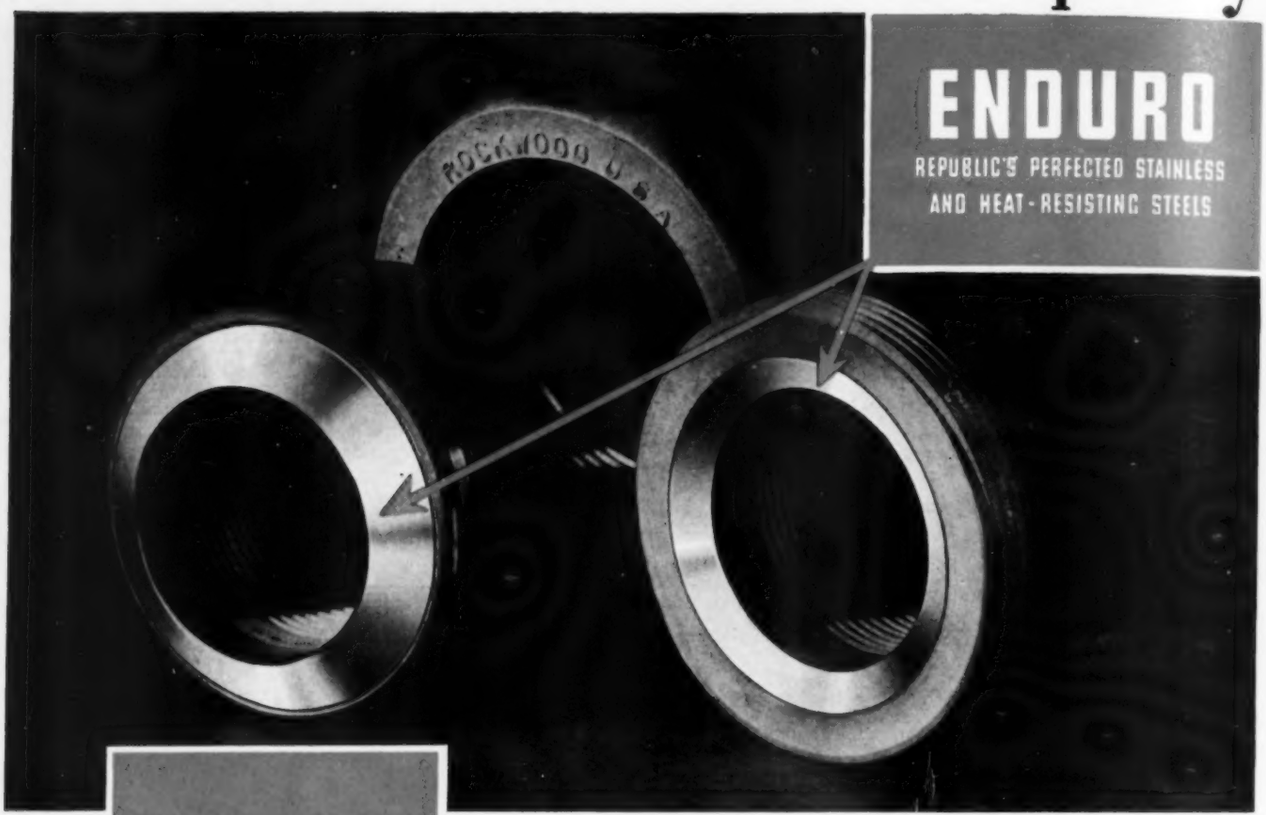

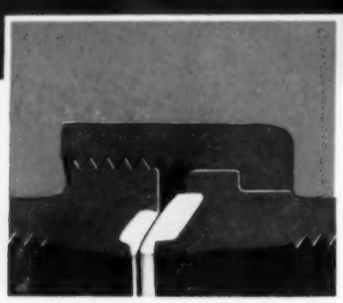

Industry moves ahead with more efficient and more economical processes to produce better products, more useful service and greater profits. But in doing so, higher working pressures and temperatures together with corrosive elements and compounds exact a heavy toll on equipment.

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THE EDITOR'S PAGE

June, 1936

?Which of the articles published this month most interests you? On what subject would you like to see more articles? The Editors would like to know, as it will be helpful in developing material for future publication, will aid them in editing H. P. & A. C. to meet your every-day needs and interests. Your cooperation in filling in and returning the coupon will be appreciated.

The Editor,
HEATING, PIPING AND AIR CONDITIONING,
Six North Michigan Avenue,
Chicago, Illinois.

The following article in the June, 1936, H. P. & A. C. most interests me.....

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■ Feature of the Air Conditioning Manufacturers' Association annual meeting last month at Hot Springs, Va., was presentation of an air conditioning applications code for the use of members pending drafting of a national code by interested technical societies; submitted by the chairman of the technical and code committee, Donald French, the code was approved by the association and will be released soon. Officers elected are: President, J. F. G. Miller, B. F. Sturtevant Co.; Vice-president, J. A. Harlan, Kelvinator Corp.; Treasurer, P. A. McKittrick, Parks-Cramer Co.; Board of Directors, S. E. Lauer, chairman, York Ice Machinery Corp., W. F. Armstrong, Delco-Frigidaire Conditioning Corp., P. Y. Danley, Westinghouse Electric & Mfg. Co., J. J. Donovan, General Electric Co., William H. Price, Jr., Carrier Engineering Corp., and the officers. . . . Walter A. Bowe has joined the Carrier Engineering Corp. as advertising manager, it has been announced by L. R. Boulware, vice-president and general manager.

■ *On May 21, the formal opening of the shops and laboratories of the Refrigeration and Air Conditioning Institute, 2150 Lawrence Ave., Chicago, was celebrated with an all day open house and dinner in the evening. Executives and representatives of leading refrigeration and air conditioning manufacturers were present to inspect the facilities of the school for training, through its home study course and shop experience, future service men, engineers, salesmen and executives for the industry. Ray D. Smith, president of the Institute, was the only scheduled speaker.*

■ The home office of the Ford Motor Co. announced last month a two year program calling for the air conditioning of the administrative offices of principal Ford branches in the United States; total estimated cost: $260,000. Ford's home offices, engineering laboratory, and the Dearborn Inn were air conditioned last year, as were key departments of the Rouge plant, including the foundry machine shop, molding rooms of the Rouge plant foundry, chemical and metallurgical laboratories, drafting rooms, ceramics laboratory, and x-ray room . . . As a further addition to Minneapolis-Honeywell Regulator Co.'s research facilities and to provide for comfort and efficiency of office employees, three complete air conditioning installations have been made at the main office and plant in Minneapolis, total capacity beig 248.5 tons. On the top of the enlarged plant, a test bungalow has been built for heating and air conditioning research . . . Registration at the Conference on Air Conditioning at the University of Illinois last month totalled 203, with persons present from eleven states and one foreign country; aim of the meeting was to present practical information to the non-technically trained citizen. The papers given are to be published under one cover by the U. of I. Engineering Experiment Station. . . . The largest air cooled and conditioned hotel convention area in the world will be provided by the Stevens Hotel, Chicago, when installations now being made are completed, according to Otto K. Eitel, manager. Supplementing previously installed equipment, work has been started on cooling four large ballrooms, six private dining rooms, and a council room.

■ *Which article in this month's* HEATING, PIPING AND AIR CONDITIONING *most interests you? On what subject would you like to see more articles? The Editors would like to know, as it will be helpful in developing material for future publication, will aid them in editing* H. P. & A. C. *to meet your ever-day needs and interests. Your cooperation in filling in and mailing the coupon on this page will be appreciated.*

■ Because the costs involved in surveying, engineering, and estimating air conditioning jobs are substantial items, increased attention is being given to methods of keeping them down. . . . C. H. Lewis of R. L. Spitzley Heating Co. pointed out in a recent issue of the Heating, Piping and Air Conditioning Contractors National Association "Official Bulletin" that failure of the sales engineer to measure the prospective purchaser on his buying interest, ability to buy, and the approximate cost of the system he requires leads to dead timber which results in high average sales and engineering costs per job sold. . . . Manufacturers as well as contractors are endeavoring to solve the problem; one has developed a "value estimate" which must be filled out before a job is engineered, has as its object to determine whether or not the prospective owner is justified in investing in air conditioning, consideration being given to his particular situation and requirements. Percentage tables showing what air conditioning has meant to other establishments in the same business are used as a basis for the forecast. . . . H. D. Betz of Betz Unit Air Cooler Co. has found it advisable to aid the prospective owner to visualize the engineering and equipment involved in a complete air conditioning system, and developed the two charts published last month for this purpose.

■ *"To define air conditioning is a comparatively simple matter, but to furnish it involves some of the most complex engineering problems industry has ever faced," said William B. Henderson, executive vice-president of the Air Conditioning Manufacturers' Association at the building owners and managers con-*

[Continued on page 96, back section]

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Published monthly by the Keeney Publishing Company, 6 N. Michigan Ave., Chicago (Telephone: State 6916). F. P. Keeney, President; W. J. Osborn, Vice-President; R. Payne Wettstein, Secretary; Chas. E. Price, Treasurer. Address all correspondence to Chicago office. New York office, Room 1950, Grand Central Terminal Bldg. (Telephone: Murray Hill 2-8293). Cleveland office, 2047 Rossmoor Road (Telephone: Yellowstone 1540). Los Angeles office. 1406 S. Grand Ave. (Telephone: Richmond 6191). Subscriptions—U. S. and possessions, Canada, Mexico, Central America, South America, $2.00 per year; other, $4.00. Current copies, U. S. and possessions, 25c each, back copies 50c each. January, 1936, issue, $1.00 per copy. Member of Audit Bureau of Circulations and The Associated Business Papers, Inc.

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Heating · Piping and Air Conditioning

Vol. 8, No. 6 • *June, 1936*

FLANGED PIPING JOINTS

Methods for Applying and Determining Bolt Stresses

By **E. C. Petrie***

Leakage at the joints is one of the most serious problems encountered in high pressure, high temperature piping systems. As one of the factors upon which the pressure retaining efficiency of a bolted flanged connection is dependent is the initial bolt load applied, investigations have been made on the methods of applying bolt loads and on the efficiency of a bolt and nut assembly. Results of the studies are given here, as is a comparison of data previously available

WHILE a flanged joint is one of the oldest means for joining two pressure sustaining units, it is still the object of much investigation, as the leakage at joints is one of the most serious problems encountered in high pressure, high temperature piping installations.

The pressure retaining efficiency of a bolted flanged connection is dependent upon many factors, among which are the initial bolt load applied, the relative flexibility of flange and bolt, the gasket material, gasket compression, and the temperature of the contained fluid. The effect of each of these factors on flange joint efficiency requires considerable investigation and each presents a problem which is entirely independent of the other with relation to the determination of individual characteristics. For example, the determination of the magnitude of the initial bolt load applied is directly dependent upon the efficiency of the bolt and nut assembly and upon the method of applying this load. It is the purpose of this article, therefore, to describe the various methods used in applying bolt loads and also to present data obtained by recent investigations on the efficiency of a bolt and nut assembly.

To simplify this article the term "bolt" is applied to all types of screw fastenings; this does not indicate that all of the units tested were bolts with one nut and an upset head. The three general types of units used in practice are:

*Engineer, Division of Research and Development, Crane Co., Chicago, Ill.

Bolt—A bolt is threaded on one end only, the other end being upset into the form of a head.

Stud—A stud is threaded on both ends, one end being oversize in diameter to fit tightly (wrench fit) in a tapped hole, the other being to the standard diameter for a nut assembly.

Bolt-Stud—A bolt-stud is threaded either on both ends, or full length to the standard diameter for nut assembly on each end.

The types of screw fastenings described in this article are the "Coarse Thread Bolt and Bolt-Stud" and the "Alloy Steel Bolt-Stud." Both are referred to as "bolts."

The material and specifications for the bolting used are as follows:

Coarse Thread Bolts

Material—Open hearth free-machining steel complying with *A. S. T. M.* Specification *A*107, open hearth screw-steel grade having the following physical properties:

	Minimum
Tensile strength, lb per sq in.	60,000
Yield point, lb per sq in.	35,000
Elongation in 2 in., per cent	25
Reduction of area, per cent	50

Nuts were hot pressed, made to the American Standard heavy hexagonal dimensions. All bolts were threaded in accordance with the American Standard screw threads, coarse-thread series (B 1.1).

Coarse Thread Bolt-Studs

Material—Cold rolled steel complying with *A. S. T. M.* Specification A108, open hearth screw-steel grade having the following physical properties:

	Minimum
Tensile strength, lb per sq in.	70,000
Yield point, lb per sq in.	60,000
Elongation in 2 in., per cent	18
Reduction of area, per cent	45

Nuts were cold punched, semi-finished, made to the American Standard heavy hexagonal dimensions. All bolt-studs were threaded in accordance with the American Standard screw threads, coarse-thread series (B 1.1).

Alloy Steel Bolt-Studs

Material—The alloy steel used in the tests described in this article complied with the requirements of *A. S. T. M.* Specification A96 for Class C bolting and the American Steel Flange Standards B16e-1932 and had the following physical properties:

	Minimum
Tensile strength, lb per sq in.	130,000
Yield point, lb per sq in.	110,000
Elongation in 2 in., per cent	18
Reduction of area, per cent	55

All these alloy steel bolt-studs were threaded full length. Nuts were medium carbon, oil quenched, semi-finished, made to the American Standard heavy hexagon nut dimensions, and met the requirements of *A. S. T. M.* Specification A96 for Class 1. All bolt-studs from ½ in. up to and including 1 in. diameter were threaded in accordance with the American Standard screw threads, coarse-thread series (B 1.1). Bolt-studs having diameters 1⅛ in. and larger were threaded with the American Standard screw threads, 8-pitch-thread series.

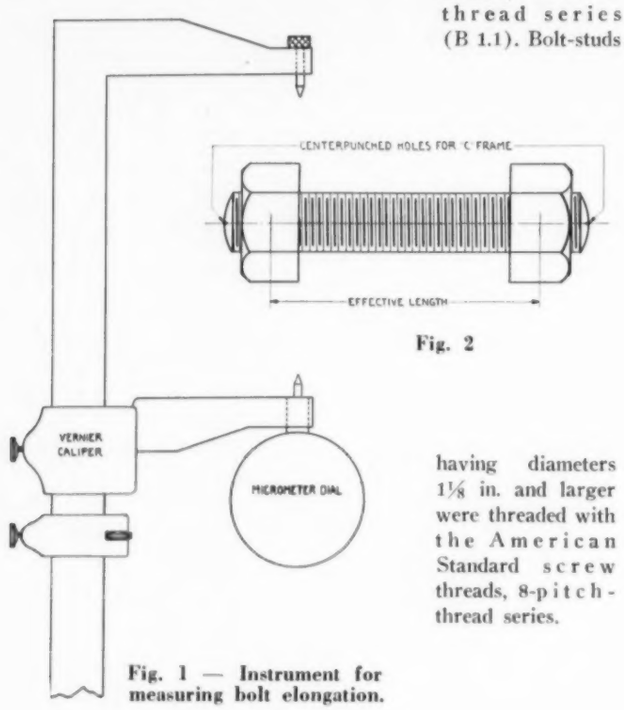


Fig. 2

Fig. 1 — Instrument for measuring bolt elongation.

Methods for Determining Bolt Loads

When a nut is tightened on a bolt, there are two types of stresses which are produced, (1) a tensile stress caused by the axial load, and (2) a torsional stress caused by the resistance of the nut and bolt threads to the turning effort. The magnitude of both of these stresses in practice is primarily dependent upon the smoothness of the threads both in the nut and on the bolt, the lubrication of the bearing surfaces, the bolt and nut materials, and the type and number of threads. It is assumed for all practical purposes that the torsional stress can be disregarded because it is generally very small in comparison with the direct tensile stress.

There are two generally accepted methods of determining bolt loads applied to flange joints, the elongation method and the torque method. The turning effort applied to the nuts in the elongation method may be obtained either by pulling or sledging up the nuts or by elongating the bolts by means of heat.

Elongation Method

When a steel bar is subjected to a direct tensile pull it elongates approximately 0.001 in. per in. of length when a stress of 29,000 lb per sq in. is applied. For example, if a steel bar having a cross-sectional area of 2 sq in. is loaded axially with a load of 58,000 lb, each inch of length will elongate 0.001 in. If the stress, 29,000 lb per sq in., is divided by the unit strain, 0.001 in., the figure 29,000,000 is obtained, which has been termed "Young's Modulus," or the "Modulus of Elasticity of Steel." Similarly, when a nut is turned on a steel bolt in a flange joint in such manner as to produce a shortening of the distance between the nut faces, a tensile load is set up in the bolt which is dependent upon the degree of tightening of the nut. If the bolt is of uniform cross section throughout its length, that is, either full threaded or turned down to the root diameter, the elongation will be based upon the effective length of the bolt and the cross sectional area at the root of the thread. If, however, the bolt is of the full shank type, that is, with a non-uniform cross section, the total elongation will be the sum of the elongations based on the length of each cross section.

It has been mentioned that in applying the loads to bolts the nuts can be turned either by pulling up or sledging up. The sledging can be performed by using a sledge hammer on a small handled wrench or by the use of an air hammer. If there is not enough room to apply either of these methods, the elongation in the bolt can be obtained by the thermal expansion method by heating the bolt to a predetermined temperature, or until it becomes elongated a certain amount, and then tightening the nuts slightly, after which the bolt is permitted to cool and contract in the flange joint. As the bolt contracts, the load between the nut and flange faces increases and the elongation of the bolt changes from a thermal expansion into an increase in original length which is a measure of the stress in the bolt as described above.

A discussion of the means of measuring elongation, the effective length of bolt to use, the necessary heat, and subsequent problems which arise, is given later under the subject of "Data for Practical Application."

Torque Method

When the nut of a bolt is turned so as to increase the bolt stress, the turning effort increases approximately in direct proportion to the stress. The torque or turning moment may be measured by means of a dynamometer or spring scale attached to a lever of known length. The torque applied is usually measured in foot-pounds, which is evaluated by multiplying the load by the length of the lever or wrench.

The resultant stress in the bolt for a definite torque varies with the type of threads, number of threads, lubrication of bearing surfaces, etc., and this method therefore gives only an approximation of the stress in the bolt. However, for all practical purposes it is possible

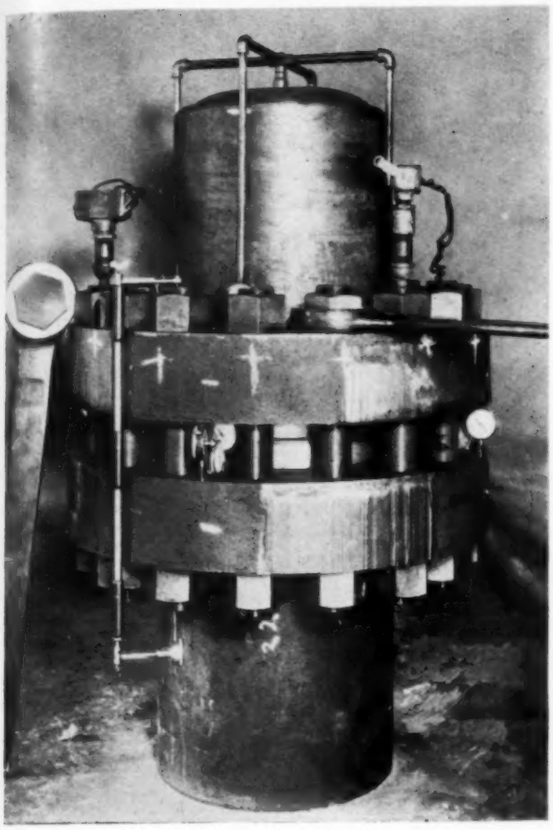


Fig. 3—Test set-up for 20 in., 1500 lb cast steel flanged joint showing heaters and instruments in place

to establish data on torque-stress relationships which should be of great benefit to those who erect piping. Recent investigations have led to certain definite recommendations which are given under the following heading.

Data for Practical Application of Bolt Loads

The necessity for accurately determining bolt stresses has been the subject of many discussions. It has been stated that when bolts are installed, enough load should be applied to make the joints tight. This is true, but in some cases the bolts may be pulled up excessively, under which condition the joint will not give satisfactory service. An example of the latter was lately brought to our attention in connection with bolting up return bends for oil stills. In this instance, excessive initial stresses were applied to the bolts, and when the pressure and temperature were introduced, the joints leaked, due to the plastic extension of the bolts caused by the higher loads developed by the temperature difference between the bolts and return bend body. The solution was to apply lower initial loads to the bolts when pulling them up in the return bends so that any additional load due to temperature difference would not cause the total stress to exceed the yield point of the bolting material.

With the higher temperatures and pressures now used in oil refineries, power plants and industrial processes, it is becoming very important that the initial bolting loads applied be known.

In order to establish definite and accurate data which can be used in determining initial bolt stresses, we instituted and carried out a series of research investigations. The major results are presented in the following paragraphs.

Application of the Elongation Method

The method of determining bolt stress by measuring the elongation of a bolt after the nut has been tightened is probably the most fundamental and accurate way of ascertaining the actual load applied. As has been described, this method is based upon the elongation of each inch of effective length and is dependent only upon the area of the root of the thread or shank and the bolt material. It is not affected by improper lubrication, poor bearing surfaces or any of the other variables which enter into the nut and bolt assembly.

Instruments and Methods Used for Measuring Elongation of Bolts

It is difficult to measure the elongation of a bolt accurately unless the proper instruments are used. The following recommendations are based upon data gleaned from our experience and tests:

1. A "C" frame spanning the entire length of the bolt and equipped with an indicator dial should be used. Fig. 1 shows an instrument of this type. This frame should be rigid in construction and should not be deformed by its own weight, and the dial should be graduated in thousandths of an inch or smaller graduations.

2. Very sharp centerpunch or drill marks should be placed in one or both ends of the bolt to supply resting places for the measuring instrument. If the mark is only in one end, the other end must be smooth to allow the rounded surface of the "C" frame point to be moved about. See Fig. 2. Flattened surfaces may give sufficiently accurate measurements but the results may be questionable. For very accurate work, small balls may be placed in each end of the bolt. The points of the "C" frame should be flat at the dial end and concave at the other end.

3. Strain gages spanning only 1 in. or so of the length of the bolt may give readings which are not indicative of the elongation of the entire strength due to bending of the bolt caused by eccentric loads.

4. The effective length of a bolt should be assumed as the distance between the centers of the two nuts. See Fig. 2. The reading obtained from the indicator dial should be divided by this length in inches to obtain the elongation of the bolt in inches per 1 in. of length.

Method of Inducing Bolt Elongation by Heat

The method of inducing bolt elongation by heat is best described by a practical example. The test as outlined below is one of the many which has been conducted.

A 20 in., 1,500 lb cast steel flanged joint equipped with sixteen 3 in. full threaded bolts was assembled in the following manner:

Longitudinal holes, 5/8 in. in diameter, were drilled directly through the center of each of the sixteen bolts. The joint was assembled and snugged up by applying approximately 350 ft lb to the nuts. Then, one bolt at a time, each successive one being located diametrically opposite to the preceding one, was loosened, supported and held in place from below by means of a screw jack, and an electric heating element was inserted into the hole drilled through the center of the bolt. The over-all length of the bolt was measured at room temperature and was checked, as the temperature of the stud was gradually increased, by means of a dial micrometer of

special construction. The points of this micrometer were set into centerpunch marks in the ends of the bolts, as shown in Fig. 3.

When the bolt was elongated sufficiently, the nuts were pulled down into intimate contact with the backs of the flanges by the application of approximately 350 ft-lb torque. (One man used a 3 in. box wrench, 46 in. in length.) The bolt was then allowed to cool and its tendency to contract was opposed by the presence of the flanges between the nuts. The flanges, however, were flexible and did spring together somewhat.

Because of this flange bending, much of the induced

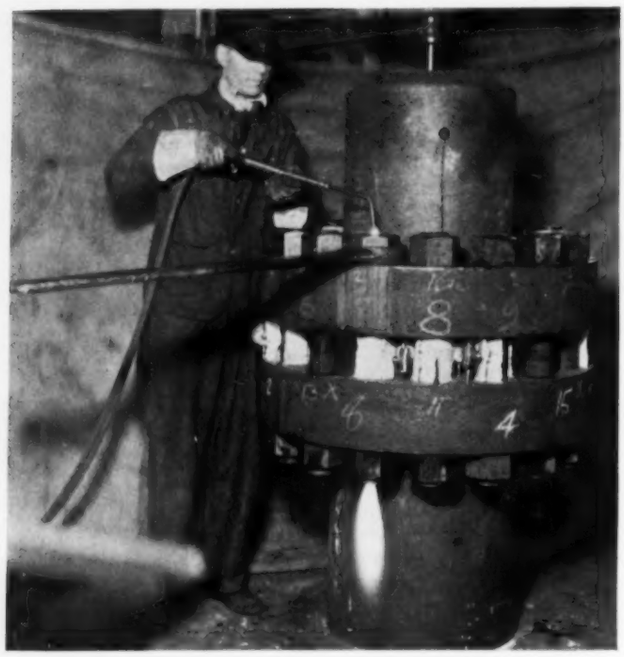


Fig. 4—Method of heating bolts with an acetylene torch

bolt elongation—roughly 50 per cent—was lost as the bolt cooled, and it was, therefore, necessary to elongate the bolt, under temperature, about two or three times as much as was required in the bolt after cooling.

In stressing these bolts, the effective length of the bolt was assumed as the distance between the centers of the two nuts, that is, each inch of this length is assumed to elongate uniformly under a given stress. Therefore, the total elongation retained in the bolt when cool, divided by the distance between the centers of the two nuts, indicates the unit elongation in inches per inch. The product of this unit elongation and the modulus of elasticity (approximately 29,000,000 for steel) is the fiber stress in lb per sq in. existing in the bolt material.

To check the bolt stress as determined by total elongation measurements, strain gages having 2 in. spans were mounted on that section of several bolts exposed between the flanges. The readings obtained by these strain gages were found to be less reliable than those obtained by measuring the total elongation of the bolt because, as the bolting load was increased, the flanges deflected, the faces of the nuts tending to remain in intimate contact with the backs of the flanges, and a bending of the bolts resulted. Because of the location of the strain gages, the effect of this bolt bending neutralized to some extent the readings which would normally be obtained under purely tensile loading.

At best, the work of pulling up a flanged joint by heating the bolts is tedious and it becomes particularly so when the accuracy of stress determination required in test work must be maintained. The yielding of the flanges and the slow process of heating the large bolts by electrical elements added to the tedium of the process, but this condition was relieved somewhat by the substitution of an acetylene flame as the heating element. See Fig. 4.

Regardless of which method was employed in heating the bolts, they were always insulated, while being heated, from the flanges and from the jack which was used to support them from below.

Application of the Torque Method

Earlier torque-stress relationship tests were conducted in 1921 on several different types of bolts, which indicated that bolts having eight threads per inch required less torque than those having a smaller number of threads (4½ to 7) to produce the same stress. These tests also indicated that the use of semi-finished case hardened nuts required less torque to produce a given stress than did a cold pressed nut. The values of the torque necessary to produce definite stresses in all sizes of bolts up to and including 2 in. were tabulated from these test data, Tables 4 and 5, Column 3.

The material for the high temperature bolts used in the 1921 tests was a chrome-nickel steel and the nuts were case hardened. As the bolts used in the present investigation were of chrome-molybdenum steel with oil quenched nuts, studies were made to determine the effect of these changes on the previously published data.

Before making the tests, a thorough study of the available literature on this subject, summarized briefly below, was made.

George S. Case[1]—When a reasonably well-lubricated bolt (coefficient of friction = 0.15) is tightened, the relation between the torque on the nut and the tensile stress in the bolt is given by the equation

$$PL = RDT$$

where P = Pull at end of wrench, lb
L = Effective length of wrench, in.
D = Nominal or outside diameter of bolt, in.
T = Load per bolt, lb
R = 0.200

See Tables 4 and 5, Column 4, for data pertaining to the torque-stress relationship as established by this formula. These data have been taken directly from the above article.

Sabin Crocker[2]—The conversion of wrench torque into tension in the bolt material is aided through the mechanical advantage furnished by the inclined plane of the screw thread, but resisted in turn by friction in the threads and in the bearing surface between the under side of the nut and the flange. For well cut threads and machined bearing surface for the nut (*i.e.*, the under side of the nut and the back of the flange machined or spotfaced) various authorities assign an efficiency of 11 to 12 per cent. If the threads and bearing surface are well lubricated the efficiency can be increased to about 18 per cent. These values refer only to the efficiency of the contact surfaces determined from the coefficient of friction and do not involve the

[1]Stresses on Bolts—Nut Dimensions—Wrench Design, *Mechanical Engineering*, September, 1926.
[2]"The Design of Flanged Pipe Joints," Heating, Piping and Air Conditioning, August, 1929.

pitch of threads, which is taken into account as explained below. The stress theoretically produced by wrench torque can be computed as follows:

$$S = \frac{2\pi eNT}{a} \quad \text{or} \quad T = \frac{Sa}{2\pi eN}$$

where

S = Stress at root of thread, lb per sq in.
e = Efficiency as described above
N = Number of threads per inch
T = Torque in in.-lb
a = Cross-sectional area at root of thread, sq in.

See Table 5, Column 5, for values determined by the use of this formula using a value for e of 12 per cent.

John J. Harman[3]—The following formula is given to determine the torque to produce definite stresses in bolts.

$$T = QS\left[\tan(a+\emptyset) + \tan\emptyset\right]$$

where

T = Torque, in.-lb
Q = Total bolt load, lb
S = Pitch radius of threads
a = Angle of thread, degrees.

$$\tan a = \frac{P}{6.28S}$$

P = Pitch of thread, inches.
$\emptyset$ = Friction angle = 10¾ degrees
$\tan\emptyset = 0.19$ = Coefficient of friction

See Table 4, Column 5, for data established by this formula. The values given have been based on Fig. 2, given in Mr. Harman's article.

Clarence W. Ham and David G. Ryan[4]—The formula given for the torque necessary to produce a definite stress in a bolt when the bolt is turning in a fixed sleeve or nut may be expressed as follows:

$$T = \frac{Qd}{2}\left[\frac{\cos\beta\tan a + u}{\cos\beta - u\tan a}\right]$$

where

T = Torque, in.-lb
Q = Load, lb
β = 30 degrees

$$\tan a = \frac{\text{Lead}}{d}, \quad \text{Lead} = \frac{1}{\text{No. of threads per inch}}$$

d = Pitch diameter
u = Coefficient of friction.

The formula for the moment of friction of a collar thrust bearing or nut is given as follows:

$$M = \frac{u'Q}{2}(r_2 + r_1)$$

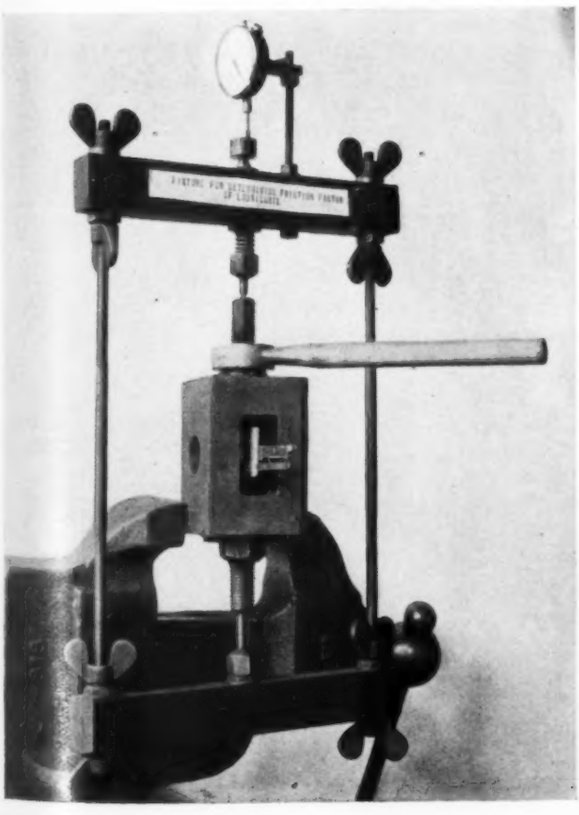
where

M = Moment of friction, in.-lb
Q = Load, lb
r_2 = Outside radius of collar
r_1 = Inside radius of collar
u' = Coefficient of friction

Assuming a coefficient of 0.16 for the formula given for thread friction and an experimentally derived factor of 0.50 for u', the values given in Tables 4 and 5, Column 6, were determined for well lubricated bolts.

[3]"Testing High Pressure Steam Piping," Heating, Piping and Air Conditioning, April, 1932.
[4]"An Experimental Investigation of the Friction of Screw Threads," University of Illinois *Bulletin* No. 247.

Fig. 5—Fixture for determining bolt stress and friction factor of lubricants



New Torque-Stress Relationship Tests

Because of the variable results obtained by the several investigators mentioned, it was decided to test bolts in sizes ⅝ in. to 2 in. inclusive to determine the torque-stress relationship under actual installation conditions.

A number of each of the following sizes of bolts made from *S.A.E.* 4140 alloy steel equipped with *S.A.E.* 1045 heat treated oil quenched nuts were installed in a special cold rolled steel hexagon block bored out to a diameter ⅛ in. larger than the diameter of the bolt.

Size of Bolt, in.	Length of Block, in.	Effective Length Center to Center of Nuts, in.
⅝	2¾	3⅜
1	5	6
1¼	10	11¼
1½	10	11½
1¾	10	11¾
1⅞	10	11⅞
2	10	12

The threads and nut bearing surfaces were well lubricated with a thin paste of graphite and 600W oil and the bottom nut was welded to the block and bolt to eliminate any torsional motion or added friction due to turning of more than one nut. A deflectometer frame with a dial graduated to 0.0001 in. was used to measure the change in length of the bolt when various torque loads were applied to the nut which was free to turn. See Fig. 5.

One ⅝ in. and one 1 in. bolt were tested under the above conditions, five trials being made on each size. Three of each of the 1¼, 1½, 1¾, 1⅞ and 2 in. sizes were tested in an identical manner except that the averages of nine trials were determined. Table 1 indicates the total average elongation of each size bolt when various torque loads were applied. Fig. 6 indicates the

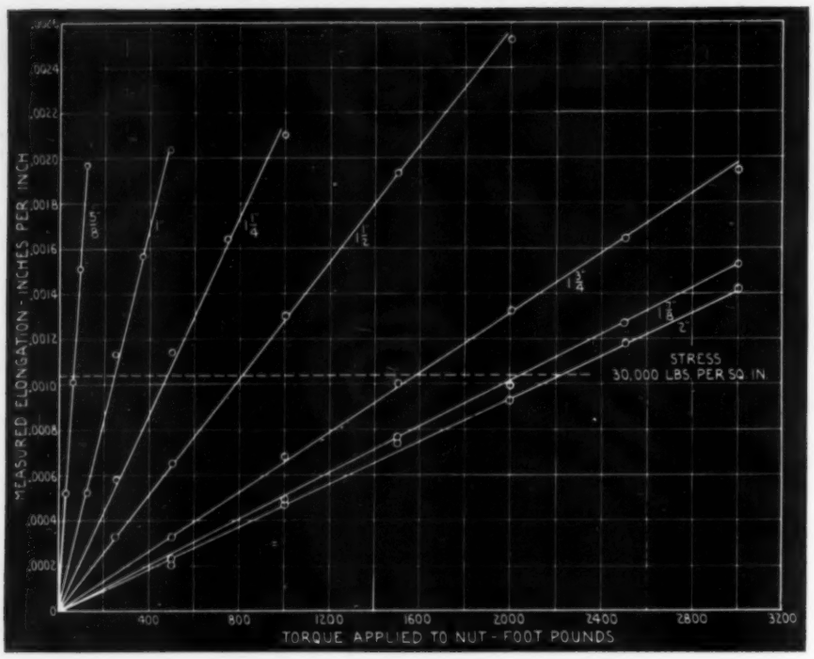


Fig. 6—Test results. Torque-strain relationship. Alloy steel bolts, ⅝ in. to 2 in. inclusive

Table 1—Total Elongation of Alloy Steel Bolts in Inches Test Data

Applied Ft-Lb	⅝ In.x 3⅜ In.	1 In.x 6 In.	1¼ In.x 11¼ In.	1½ In.x 11½ In.	1¾ In.x 11¾ In.	1⅞ In.x 11⅞ In.	2 In.x 12 In.
30	0.00175						
60	0.0034						
90	0.0051						
120	0.00665						
123		0.0031					
245		0.0068					
250			0.0065	0.0038			
367		0.0094					
490		0.0122					
500			0.0128	0.0075	0.0039	0.0027	0.0025
750			0.0184				
1000			0.0236	0.0150	0.0080	0.00585	0.0058
1500				0.0222	0.0118	0.0091	0.0089
2000				0.0289	0.0155	0.0118	0.0111
2500					0.0193	0.015	0.0142
3000					0.0228	0.0181	0.017

Note—Bolt size given as nominal size of bolt and effective length of bolt, center to center of nuts.

Table 2—Torque-Stress Relationship for Alloy Steel Bolts Load in Pounds on Bolts When Torque Loads Are Applied

Nom. Dia. of Bolt, Inch	No. of Th'ds per Inch	Dia. at Root of Thrd.	Area at Root of Thrd.	Stress: 30,000 Lb per Sq In — Torque Ft-Lb	30,000 Lb per Sq In — Compression Lb	45,000 Lb per Sq In — Torque Ft-Lb	45,000 Lb per Sq In — Compression Lb	60,000 Lb per Sq In — Torque Ft-Lb	60,000 Lb per Sq In — Compression Lb
¼	20	0.185	0.027	4	810	6	1215	8	1620
5/16	18	0.240	0.045	8	1350	12	2025	16	2700
⅜	16	0.294	0.068	12	2040	18	3060	24	4080
7/16	14	0.345	0.093	20	2790	30	4185	40	5580
½	13	0.400	0.126	30	3780	45	5670	60	7560
9/16	12	0.454	0.162	45	4860	68	7290	90	9720
⅝	11	0.507	0.202	60	6060	90	9090	120	12120
¾	10	0.620	0.302	100	9060	150	13590	200	18120
⅞	9	0.731	0.419	160	12570	240	18855	320	25140
1	8	0.838	0.551	245	16530	368	24795	490	33060
1⅛	8	0.963	0.728	355	21840	533	32760	710	43680
1¼	8	1.088	0.929	500	27870	750	41805	1000	55740
1⅜	8	1.213	1.155	680	34650	1020	51975	1360	69300
1½	8	1.338	1.405	800	42150	1200	63225	1600	84300
1⅝	8	1.463	1.680	1100	50400	1650	75600	2200	100800
1¾	8	1.588	1.980	1500	59400	2250	89100	3000	118800
1⅞	8	1.713	2.304	2000	69120	3000	103680	4000	138240
2	8	1.838	2.652	2200	79560	3300	119340	4400	159120
2¼	8	2.088	3.423	3180	102690	4770	154035	6360	205380
2½	8	2.338	4.292	4400	128760	6600	193140	8800	257520
2¾	8	2.638	5.259	5920	157770	8880	236655	11840	315540
3	8	2.838	6.324	7720	189720	11580	284580	15440	379440

Table 3—Torque-Stress Relationship for Coarse Thread Bolts Load in Pounds on Bolts When Torque Loads Are Applied

Nom. Dia. of Bolt, Inch	No. of Th'ds per Inch	Dia. at Root of Thrd.	Area at Root of Thrd.	Stress: 7,500 Lb per Sq In — Torque Ft-Lb	7,500 Lb per Sq In — Compression Lb	15,000 Lb per Sq In — Torque Ft-Lb	15,000 Lb per Sq In — Compression Lb	30,000 Lb per Sq In — Torque Ft-Lb	30,000 Lb per Sq In — Compression Lb
¼	20	0.185	0.027	1	203	2	405	4	810
5/16	18	0.240	0.045	2	388	4	675	8	1350
⅜	16	0.294	0.068	3	510	6	1020	12	2040
7/16	14	0.345	0.093	5	698	10	1395	20	2790
½	13	0.400	0.126	8	945	15	1890	30	3780
9/16	12	0.454	0.162	12	1215	23	2430	45	4860
⅝	11	0.507	0.202	15	1515	30	3030	60	6060
¾	10	0.620	0.302	25	2265	50	4530	100	9060
⅞	9	0.731	0.419	40	3143	80	6285	160	12570
1	8	0.838	0.551	62	4133	123	8265	245	16530
1⅛	7	0.939	0.693	98	5190	195	10380	390	20760
1¼	7	1.064	0.890	137	6675	273	13350	545	26700
1⅜	6	1.158	1.054	183	7905	365	15810	730	31620
1½	6	1.283	1.294	219	9705	437	19410	875	38820
1⅝	5½	1.389	1.515	300	11363	600	22725	1200	45450
1¾	5	1.490	1.744	390	13080	775	26160	1550	52320
1⅞	5	1.615	2.049	525	15368	1050	30735	2100	61470
2	4½	1.711	2.300	563	17250	1125	34500	2250	69000

Table 4—Foot-Pounds to Produce 30,000 Lb Per Sq In. Stress in Alloy Steel Bolts

Size of Bolt, Inch	No. of Thrds. per Inch	1921 Values	Case	Harman	Calculated from Univ. of Ill. Formula	New Adopted Values
¼	20	4			4	4
5/16	18	8			7	8
⅜	16	14			13	12
7/16	14	23			19	20
½	13	33		33	29	30
9/16	12	55			42	45
⅝	11	70		64	58	60
¾	10	90		110	100	100
⅞	9	160		190	163	160
1	8	280		280	244	245
1⅛	8		360	400	355	355
1¼	8		425	550	498	500
1⅜	8		715	760	680	680
1½	8		1025	1000	893	800
1⅝	8		1140	1300	1113	1100
1¾	8		1560	1550	1456	1500
1⅞	8		1870	1900	1750	2000
2	8		2520	2500	2204	2200
2¼	8		3266	3300	3180	3180
2½	8		4535		4400	4400
2¾	8		6142		5920	5920
3	8		8063		7720	7720

average elongation per inch of length for all sizes and was plotted from the data given in Table 1.

From the data given in these tests, Fig. 7 was prepared; it indicates the straight line logarithmic relationship of the test values, as shown by the solid squares, for 30,000 lb per sq in. stress. The circled points indicate the adopted values of the torque necessary to produce a stress of 30,000 lb per sq in. in bolts with threads as given in Table 2. The solid circles indicate the adopted values for bolts with threads as given in Table 3.

Table 2 lists the torques to produce stresses of 30,000, 45,000, and 60,000 lb per sq in. in well lubricated alloy steel bolts.

Table 3 lists the torques to produce stresses of 7,500, 15,000, and 30,000 lb per sq. in. in well lubricated coarse thread bolts.

Table 4 shows the comparison of the original values established by the 1921 studies, results of several investigators, and the values determined by the present tests for alloy steel bolts.

Table 5 shows the comparison of the original values established by the 1921 studies, results of several investigators, and the values adopted for coarse thread bolts.

Effect of Lubrication on Torque-Stress Relationship

Tests were conducted on 1 in. alloy steel bolts having eight 60 deg "V" threads per inch, to determine the effect of various lubricants on the torque-stress relationship. See Fig. 5. The results of this test were:

Lubricant	Torque to Produce 30,000 Lb Per Sq In. Stress—Ft-Lb
No lubricant	410
Glycerine & aluminum	410
Cup grease	255
Graphite & 600W oil	255
High temperature thread lubricant	205

Effect of Thread Form on Torque-Stress Relation

Tests were also conducted on 1 in. alloy steel bolts having Acme (29 deg) threads and National Form (60 deg V) threads to determine the effect of thread profile on the torque-stress relationship. High temperature thread lubricant was used in both tests. The results were as follows:

Type of Thread	Torque to Produce Stress of 30,000 Lb Per Sq In.
60° V	245 Ft-Lb
Acme	205 Ft-Lb

Acme threads offer a greater mechanical advantage than the 60 deg "V" National Form Threads, as 19 per cent greater bolt loads were obtained with the former with a given torque load on the nuts. The difference in resultant bolt load will probably not be as great in actual practice, due to variations in thread contact surfaces and mating angles.

Table 5—Foot-Pounds to Produce 15,000 Lb Per Sq In. Stress in Coarse Thread Bolts

Size of Bolt, Inch	No. of Thrds. per Inch	1921 Values	Case	Crocker	Calculated from Univ. of Ill. Formula	New Adopted Values
¼	20	2	2	2	2	2
5/16	18	4	4	5	4	4
3/8	16	7	7	9	6	6
7/16	14	12	11	14	10	10
½	13	17	16	19	15	15
9/16	12	28	24	27	21	23
5/8	11	35	32	37	29	30
¾	10	45	57	60	50	50
7/8	9	80	91	93	82	80
1	8	140	137	138	122	123
1¼	7	220	194	197		195
1½	7	280	273	253		273
1¾	6	440		349		365
1½	6	543	473	428		437
1⅝	5½	770		546		600
1¾	5	810	745	692		775
1⅞	5	1135		813		1050
2	4½	1283	1125	1015		1125

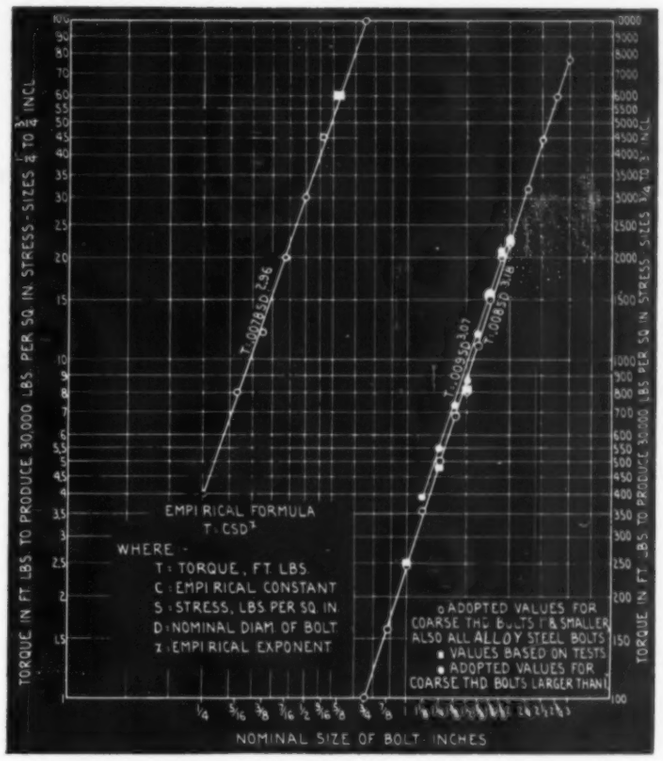

TORQUE IN FT. LBS. TO PRODUCE 30,000 LBS. PER SQ. IN. STRESS - SIZES ¼ TO 2" INCL.
TORQUE IN FT. LBS. TO PRODUCE 30,000 LBS. PER SQ. IN. STRESS - SIZES 2¼ TO 3" INCL.
NOMINAL SIZE OF BOLT - INCHES
EMPIRICAL FORMULA
T = CSD^z
WHERE -
T = TORQUE, FT LBS
C = EMPIRICAL CONSTANT
S = STRESS, LBS PER SQ IN
D = NOMINAL DIAM OF BOLT
z = EMPIRICAL EXPONENT
T = .0078 SD^2.96
T = .0085 SD^3.07
T = .0085 SD^3.18
o ADOPTED VALUES FOR COARSE THD BOLTS 1" & SMALLER ALSO ALL ALLOY STEEL BOLTS
• VALUES BASED ON TESTS
• ADOPTED VALUES FOR COARSE THD BOLTS LARGER THAN 1"

Fig. 7—Torque-size relationship. Coarse thread and alloy steel bolts

Acknowledgments

The writer gratefully acknowledges the suggestions, cooperation, and also the work in obtaining experimental data offered by his associates, J. P. Magos, E. R. Seabloom, F. R. Venton, I. H. Carlson, A. C. Grubbe, T. D. Kelly, and W. P. Kliment.[5]

[5]Of the Division of Research and Development, Crane Co.

Number of Air Conditioning Installations in Indianapolis and Vicinity

According to the Indianapolis Power & Light Co. there were 80 air conditioning installations totalling 3463½ horsepower in Indianapolis and vicinity as of April 15, 1936. Six of these jobs totalling 56¼ horsepower were installed during 1936, 39 totalling 1370¾ horsepower were 1935 jobs, and the others were put in previous to 1935.

The 1936 jobs comprise one installation each for process, a private office, a restaurant and bar, a clothing store, a jewelry store and a library. Of the total jobs (1936 and previous to 1936) 21 of the 80 are for residences, and these installations account for 46½ of the total of 3463½ horsepower. Eighteen installations and 289¾ horsepower are for restaurants and bars, 8 installations and 11 horsepower for private offices, and 7 installations and 1668½ horsepower for department stores.

Early Comfort Cooling Plants

By **G. Richard Ohmes*** *and* **Arthur K. Ohmes†**

AS EARLY as 1874 articles appeared in U. S. and European technical magazines describing attempts at air cooling. For example, Ph. Delahaye described at a meeting of the Institution of Civil Engineers in that year a cooling plant which consisted principally of an inclined, perforated partition, over which was discharged a continuous stream of cold water pumped from a nearby brook. Fresh air was blown through the partition by a fan and in this way 35,300 cfh of air was cooled from 70 to 56 deg with 264 gpm of water. A patent on a machine for purifying, drying and cooling or warming air was taken out in 1873 by A. S. Lyman; this consisted substantially of a bed of charcoal over which water was sprinkled opposite the direction of air flow, the theory being that the contact of water and air was thus a maximum.

A unique method was used to cool the Court Theater, Vienna, about 1900. An underground passage was divided longitudinally by a vertical wall. In one passage 276 zinc trays, over which water was discharged, took care of the cooling (although normally the natural coolness of the subterranean passage sufficed) and in the other there was no cooling device whatsoever. Both air currents united at the end of the tunnel to be discharged wherever desired. In principle this was a bypass system.

In 1902 an article on comfort cooling by Frederick Breithut appeared in the New York *Evening Journal*, one of the earliest attempts to stimulate general public interest in the idea of cooling hospitals and homes. Said the author—

"For more than fifty years Prof. R. Ogden Doremus, of the College of the City of New York, has been fighting for the cooling of hospitals in summer. He has done this in lectures to his students, lectures to the public, and in numerous magazine articles. In the *North American Review* for May, 1893, he has this to say: 'For many years I have urged that liquefied gases should be utilized to temper the heat of summer in our homes and hospitals, our banks and business offices, our sumptuous edifices for pleasure or piety. I have ventured to say: 'If they can cool dead hogs in Chicago, why not live bulls and bears in the New York Stock Exchange?' Citizens would be happier, merchants would make more money and physicians would save more lives with the temperature at 70 F than at 90 or 100 F.'

"These words are commencing to produce a little effect. In the new stock exchange building they have installed a system of artificial cooling by means of liquefied ammonia. The work is not yet completed, but there can be no doubt of the success of the system when it is finished.

"The Hanover National Bank, at the suggestion of Alfred Wolff, consulting engineer, placed a similar system in operation on May 15 of this year. Since then it has been operated with unqualified success. They have 12,000 ft of pipes filled with brine. These pipes are cooled by the evaporation of ammonia. They lead into receptacles, where they coil around and around. The air about them is cooled and blown upstairs by means of large fans. The temperature can be regulated just as you regulate the heat produced by a radiator in winter. They usually maintain the temperature at about 70 F.

"If this can be done in a bank, why can it not be done in our hospitals? In many medical colleges the dead bodies are kept cold by this means. Is it not more important that the living sufferers in our hospitals should be cooled? There ought to be such vigorous demand for this on the part of the public that the next legislature would have to pass a bill compelling the saving of lives by this means."

Evidently Professor Doremus had conceived the idea of cooling hospitals somewhere back in 1845, if Mr. Breithut's article is correct. Unquestionably, the idea of comfort cooling is much older, but it may be permissible to assume that the first scientific consideration of it began about that time.

The restaurant fraternity takes honors for early comfort cooling. Mr. Breithut speaks in an unquoted paragraph of an ice cream man who cooled a dining room in a hotel which he built in Staten Island, N. Y. The cooling of the air supply was accomplished by running air through metal pipes imbedded in ice and salt. This was done about 1880. In 1883, at the Hygiene Exhibition in Berlin, a restaurant was cooled by blowing air over ice. Results did not come up to expectations. Evidently, a temperature low enough could not be maintained in the room because of a leaky roof, which was cheaply built for exhibition purposes.

In 1889, Mr. Wolff designed the Carnegie Music Hall in New York. Ice was put in large racks over which the air circulated and was thus cooled. According to the building's dedication records—

"Fresh air, at any temperature desired, in large volume but at a low velocity, is introduced, and the vitiated air is exhausted. Generally, the fresh (warmed or cooled) air enters through perforations in or near the ceilings, and the exhaust is effected through registers or perforated risers in or near the floors and, passing through an elaborate system of ducts, worked into the construction of the building, is expelled above the roof. Thus, the entire audience, whether seated in the upper gallery or in the parquet, are made comfortable and supplied with an abundance of fresh air.

"The supply of air is taken through a large vertical shaft, extending from above the roof, to the heat chamber or the ice chamber in the basement, the inlet of this shaft being remote from the chimney and the exhaust shaft. The heating surface, aggregating about 15,000 sq ft, is situated near the base of the fresh air shaft, and, together with the necessary machinery, is supplied with steam from three water-tube boilers, which develop 500 hp. Through this heating surface, or at will through the ice racks, the air is drawn by four powerful blowers, each 12 ft

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high, and forced through the system of fresh air ducts into the various parts of the building. In the exhaust chamber, immediately under the roof, are situated three exhaust fans, each 10½ ft high, and the exhaust from the kitchen is accomplished by means of another. Each of the four blowers and each of the four exhaust fans is driven by a separate steam engine. The exhaust of these eight engines, and of the four electric light engines, and of the five steam pumps, is introduced into the heating system, thus economizing to a great extent the consumption of fuel.

"Although an immense volume of air is introduced, the great area of the ducts and openings makes possible its introduction at an extremely low velocity, thus avoiding objectionable drafts, so common in large auditoriums. The heating surface and other appliances are so subdivided that atmospheric changes can be immediately compensated for, and the temperature of the air introduced suited to the winter weather or the heat of summer."

The earliest attempt to replace the use of ice for comfort cooling by a refrigerating machine seems to have been made (according to a paper read before German heating and ventilating engineers at their congress in Munich in 1898) in a private home in Frankfurt. It was reported that the plant had been operating very satisfactorily for five years, which would place its design in 1892 and its installation and operation about 1893. The paper was illustrated by a drawing not reproduced in the minutes, but from the description it appears that a cooling coil was placed in an insulated box in the attic and the cooled air dropped by gravity through shafts and many small openings into the rooms. Recirculation of air, although extensively used at that time in slaughter houses, for drying hops, for the storage of fruit, etc., was considered and discarded. Fresh air was considered more important than cooler spaces.

Cooling was accomplished by direct ammonia expansion coils. The compressor had a cylinder of 4.7 in. diameter and 4.7 in. stroke and was driven by a 3 hp motor. The simplicity of the cooling scheme was highly praised, and, as a matter of fact, the operation of the system was placed entirely in the hands of the cook (in this case a woman), who assumed the role of operating engineer with ease. Tabulations show clearly the advantage of operating the machine at a high back pressure; ammonia evaporated at a temperature of between 14 and 17.6 F. The cooling coil casing, easily removable for the inspection and cleaning of the coils, was heavily insulated with cork.

The hope was expressed that, since it was shown how easily a house could be cooled, it would come into general use throughout the world, but particularly in tropical climates. The ensuing discussions were too long to give here, but the desirability of substituting well water cooling in place of low temperature machines was vigorously contested. It was also admitted that only a veritable Croesus could afford the luxury of comfort cooling.

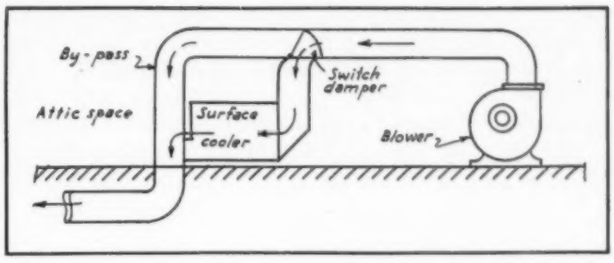


Fig. 1—Cooling system for dissecting room designed in 1899

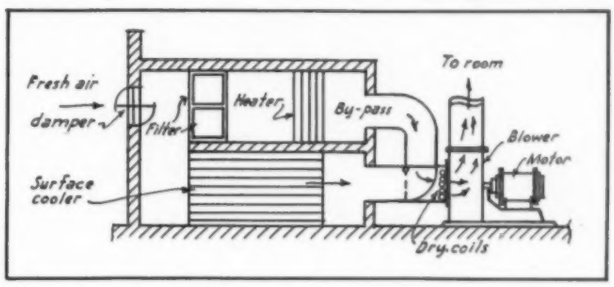


Fig. 2—Hanover National Bank installation, 1902

Apparently, the numerous discussions and articles on comfort cooling were responsible for information on the subject being included in various text books on heating and ventilation. One of the most advanced of these was "Leitfaden zum Berechnen und Entwerfen von Lüftungs — und Heizungsanlagen"[1] by Hermann Reitschel, Professor at the Royal Institute of Technology, Berlin, published in 1894 by Springer. The seventeenth chapter of this book was entitled, "Cooling of Rooms" and embodied many of the principles of air conditioning. The author gave information on the determination of heat gains to rooms, solar heat, and heat generated by people and lighting. He said that as air, through cooling, increases its relative humidity, it becomes necessary to condense moisture out of it by passing the air over low temperature surfaces and then to mix it with uncooled air, and that it was often necessary to reheat the air with heating equipment to avoid an unduly low temperature. To reduce operating costs, he stated that provisions for recirculation of air were necessary, and that this was more important when cooling than when heating.

In 1899, Mr. Wolff designed a cooling system for the post-graduate dissecting room of the Cornell Medical College in New York City, as described in the *Engineering Record*, October 5, 1901. Direct brine cooling coils were placed in bunkers in the attic space over this room and cooled fresh air (as low as 38 F) dropped by gravity into the room. The cooling coil for air supply was provided

The ever-increasing applications and public consciousness of air conditioning arouse interest in early developments in this field. So rapid has been its progress in recent years, many think of air conditioning as a product of the last decade, forget its beginnings at the turn of the century and before . . . . . This "father and son" article describes some of the early comfort cooling installations

[1] "Guide for the Calculation and Design of Ventilating and Heating Installations."

with a bypass so as to give either cooled and dehumidified air or uncooled air as shown in Fig. 1. The brine coil arrangement at the Hanover National Bank, mentioned above, is shown in Fig. 2. The bypass arrangement allowed reheating of the cooled and dehumidified air by uncooled air. An ammonia absorption machine was used to cool the brine.

Shortly after the system for the Hanover National Bank was installed, Mr. Wolff designed the large plant for the New York Stock Exchange, described in the *Engineering Record* of April 8, 1905. This plant, which consisted of three ammonia absorption machines for cooling brine, operated successfully for 18 years and was replaced in 1922 by a new and larger compression ammonia plant. The former machine had a capacity of 450 tons when using high pressure steam and 375 tons when using exhaust steam. The new plant had a nominal capacity of 750 tons, but an actual capacity of well over 1000 tons, since the large brine cooling coils permitted carrying a high evaporating pressure.

The operating conditions were carefully observed and an effort was made to determine from the large number of people who occupied the cooled spaces what temperatures would be satisfactory to the majority. The results were discussed by one of the authors—Arthur K. Ohmes—in a paper on ventilation of hospitals read before the American Medical Association in 1914—

For over ten years an artificial cooling, humidifying and ventilating plant has been in operation in the New York Stock Exchange, where 1,500 persons are busily engaged with closed windows, as is necessary where cooling is desired. Of course, this room is not cooled to the constant winter temperature of 68 F. The room temperature best liked by the average person seems to bear a fixed relation to the outside temperature and the vapor content. The most suitable inside temperatures with corresponding outside temperatures at about 60 per cent humidity are as follows:

68 outside	68 inside
75 outside	72 inside
80 outside	74 inside
85 outside	76 inside
90 outside	78 inside

At about the same time that the plant was installed in the New York Stock Exchange, a theater in Cologne was air conditioned; the plant was described in the *Gesundtheits Ingenieur*, March 10, 1904. The cooling system, installed in 1903, provided for coils to cool 47,000 cfm of fresh air. A steam driven ammonia machine of about 16 tons capacity operated 15 to 16 hr per day and stored the cold in a brine storage tank, and the stored-up cooling effect was used during the performance of three to four hours. Preliminary cooling was accomplished by well water cooling coils, which water was then used for the condensers of the refrigerating machine, and the final cooling by the brine coils.

A description of a plant for cooling a safe deposit vault of the Carnegie Safe Deposit Co., which was installed in the North Trinity Building, 115 Broadway, New York City, appeared in the *American Architect* of October 14, 1908. Brine coils were used to cool the air and the brine was cooled by an absorption machine.

A cooling system designed by Mr. Wolff in 1907 for the Metropolitan Museum of Art in New York City is particularly interesting, because it is possibly the first to attempt to maintain a constant relative humidity, so important for the preservation of paintings, wood carvings and so forth, in winter and summer. The layout

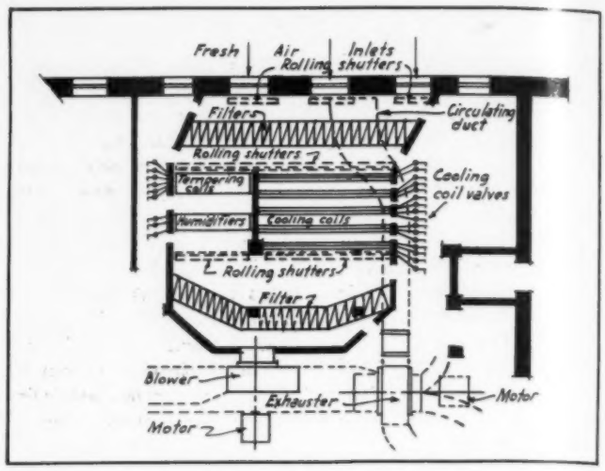


Fig. 3—Metropolitan Museum of Art job, 1907

is illustrated in Fig. 3. By manipulating dampers and rolling shutters, it is possible to maintain the following conditions of operation: (1) Fresh air supply, either warm, moistened or cooled. (2) Exhaust air only; air discharged above roof. (3) Recirculating air by discharging exhausted air into fresh air chamber. (4) A mixture of fresh air and recirculated air either warmed, moistened or cooled. (5) Bypassing cooling coils to reheat cooled and dehumidified fresh air or recirculated air or a mixture of both. (6) Bypassing tempering coils to effect quick air temperature changes in winter of air warmed by the tempering coils.

These cooling coils were used only a few years because the maintenance of about 62 F in winter with about 60 per cent relative humidity, preserved the art objects perfectly, without the use of cooling and dehumidification in the summer.

Temperature and Humidity Requirements for Air Conditioning the Meat Plant

Temperatures and humidities most suitable for air conditioning the various departments of the meat plant as given by *The National Provisioner* and reproduced by permission are:

Department	Temperatures, Deg Fahr	Relative Humidity, Per Cent
Killing Floor	*85 to 65	*60 to 20
Carcass Chill Room	32 to 40	90
Carcass Holding Cooler	34 to 36	85 to 90
Cutting Floor	55 to 60	40 to 35
Curing Cellar	34 to 40	80 to 85
Sharp Freezer	—10 to —50	95 to 100
Storage Freezer	10 to 20	85 to 100
Smoked Meat Hanging Room	60 to 65	80 to 85
Sausage Trimming Room	35 to 38	80 to 85
Sausage Meat Cooler	35 to 38	80 to 85
Sausage Mfg. Room	55 to 60	40 to 35
Sausage Prechill Room	40 to 42	90 to 95
Sausage Storage and Packing Cooler	*50 to 40	85
Offal Cooler	34 to 36	90 to 95
Cooked Ham Cooler	40 to 45	80 to 85
Bacon Slicing Room	*55 to 35	20 to 40
Assembly and Shipping Room	40 to 55	85 to 50
Loading Dock	35 to 55	95 to 55

*Range from summer to winter.

PRACTICAL HEATING

By **Harold J. Taylor***

Trouble-Shooting on Indirect Heater Jobs

Instructions for the installation of indirect hot water heaters for supplying domestic service water were given in my article on "Practical Heating" in the November, 1935, HEATING, PIPING AND AIR CONDITIONING. While this is an excellent means of heating service water, troubles result—as they do with any equipment—if selection or installation is incorrect; the present article is an attempt to prescribe the cure for those troubles which come up most frequently.

Here's another of Harold J. Taylor's discussions on practical heating problems. Subjects covered in other articles are . . . Installing Radiators Below Steam Boiler Water Line, July, 1935 issue; Adding Boiler Capacity, August, 1935; Heating Service Water with Cast Iron Boiler, September, 1935; Installing Check Valves on Radiators, and Blow-off Tanks for High Pressure Boilers, October, 1935; Installing Indirect Hot Water Heaters, November, 1935; Boiler Ratings, March, 1936; Unit Heaters, April, 1936

If the heater fails to heat, check the following:

(1) Is heater correctly connected? If heater is installed in the live steam space above the boiler water line, then the entire heater must be above the water line; I suggest in this case that the bottom of the heater be at least 20 in. above the water line.

(2) Is there an air venting valve at the bottom of the heater to vent the air from it, and is valve clean and functioning properly? This vent must be installed on the return steam piping from the heater and should be extended up 24 in.

(3) In some cases where a heater is above the water line, trouble is caused by condensation being held up in the heater and the water is not heated satisfactorily; this condition is apt to exist where the supply to the heater is much smaller than the shell openings in the heater. The small amount of steam supplied to the heater is condensed at a very rapid rate and either a vacuum is formed in the heater, holding up the condensation, or the pressure loss through the heater is so great that the condensation cannot flow to the boiler by gravity. To remedy this, increase the piping sizes. If there is an automatic regulating valve in the supply line from the boiler to the heater it is necessary to install a vacuum breaking valve in the return line from the heater.

(4) If it is wished to throttle the supply of hot water from the heater to the storage tank, a gate valve either in the supply or return piping between the heater and the tank should be used. Do not throttle the steam supply from the boiler to the heater. There are on the market automatic devices to take the place of gate valves and these may be had at nominal cost.

(5) If heater is below the water line of the boiler, be sure that both supply and return piping is below the water line. I have found jobs where the owner complained about the heater delivering little if any hot water, and on observation found that a very low water line was being carried and the tappings into which the heater was connected were partially above the boiler water line.

(6) Is heater of ample capacity? Check size of heater for the water load required and not on the size of the storage tank installed. On oil fired jobs a heater of much greater capacity should be used, as the burner may be on intermittently and the water in the boiler has a tendency to cool. Check with the manufacturer for the size of heater to use for a given job instead of guessing.

(7) Is coil inside of heater collapsed?

(8) In many cases the shell or coil openings, or both, are bushed. This is not good practice. If a heater is to deliver its full rated capacity, a required amount of boiler water must pass through it and likewise sufficient service water must circulate from the heater to the storage tank. The manufacturer has determined the tapping sizes, and they should be followed.

(9) When an indirect heater is installed with an independent gas or coal fired hot water heater which will be used only in the summer months, then install a gate valve in the return or supply line from the indirect heater to the storage tank to prevent heating of indirect heater during the summer operation, which would in turn tend to heat the water in the steam boiler.

(10) Is inside of heater shell clogged up with an accumulation of grease, sludge or lime? This must be scraped out and the heater washed clean. Where the water is hard, the inside of the heater coil will plug up with lime deposits which in many instances may be removed by muriatic acid, either by disconnecting the supply and return piping from the heater coil and pouring, little by little, acid into the upper coil opening or by removing the coil from the heater shell and submerging it in the acid. I believe it is easier to use the first method. The acid will work its way to the outlet connection, dissolving the lime. After the opening through the lime is large enough, plug up the outlet connection and fill the coil as full as possible with the acid and let it soak for 24 hours. After all the lime is dissolved, thoroughly wash out the coil and connect to the tank connections. No doubt the copper coil is damaged somewhat by the action of the acid but I have known of coils that have been cleaned several times

*Contracting Engineer, Detroit, Mich.

without appreciable wear and tear. Replacement coils may be purchased for most makes of heaters at nominal cost.

(11) The top of the heater should be as high as water line will permit, that is about 3 in. below the normal operating water line level. This will keep the heater in the hottest part of the steam boiler water.

(12) It is not good practice to install an indirect heater on a hot water heating boiler unless a boiler temperature of at least 150 F is carried, as otherwise the heat transfer from the boiler to the heater may not be satisfactory. During the coldest part of the heating season, operation may be satisfactory but during the fall and spring periods when a low boiler temperature is carried insufficient hot water will be the result. On hot water jobs, the heater supplying opening must be connected to one of the vertical radiator supply flow openings and must be installed at a point above top of boiler if best results are to be obtained. The return connection of the heater is connected into the return piping of the heating system as with a steam boiler hook-up.

(13) When a heater is connected to a horizontal storage tank, its top must be below the bottom of the tank if proper circulation and storage of hot water is to be expected. When confronted with a low installation, consider either the raising of the storage tank (computing cost of doing so) or of installing a circulating pump. If a pump is to be used, lower the top of the indirect heater so that it will be below the bottom of tank but be sure that the supply from the boiler to the heater is taken off the boiler at a point approximating 3 in. below normal water line operating level. The pump is installed between the return connection of the heater and the boiler. In this case an aquastat control is installed near bottom of storage tank and is connected electrically to the motor on the pump, which in turn causes the pump to run only when there is insufficient hot water in the storage tank.

(14) While on the subject of circulating pumps, if difficulty is experienced in getting circulation between the boiler and heater or between the heater and storage tank or both and piping, heater size, installation, etc., are apparently in good order, I suggest the use of one or two pumps to solve the problem. Manufacturers will furnish data in regard to this type of equipment and installation. With the use of a pump, the piping sizes may be considerably reduced and the heater capacity is increased approximately 25 per cent.

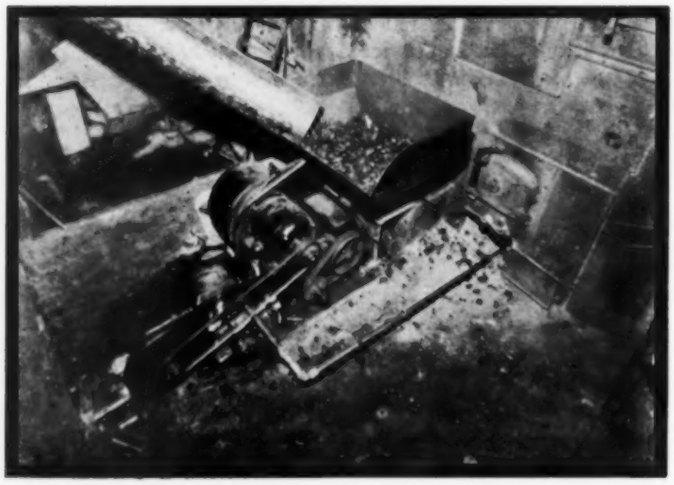
(15) On sectional boilers it is general practice to tap all sections and these tappings are connected into a manifold or header with steel or brass pipe or tubing. If we were to connect from each of the boiler sections using 1 in. connections tying into a 3 in. header, it is obvious that we would have a pocket of air extending the entire length of the header. If this is the case, then the capacity of the header is greatly reduced. To eliminate this air binding, extend a ¾ in. line from the top of the header to a point in the steam space of the water column, below the pop valve or damper regulator or other suitable opening. If the manifold or header is below the points where the 1 in. pieces tie into the boiler sections then the header will vent itself into the steam boiler, and the vent line is not required.

(16) Occasionally you will find the supply line from a heater to the tank air bound because the piping runs to a point above the tank and then drops to it. This is all right if the supply to the fixtures is taken off where the piping drops to the tank, which would vent the air from this line when hot water is drawn. On the other hand, if the supply to the fixtures is taken off another tank opening, air binding will be the result. It is advisable to change the piping connection, but if this cannot be done, an automatic air venting valve should be installed at the high point in the line.

(17) Street ells and street 45 degree ells are often used; this is poor practice. The opening through an elbow of this type is much smaller than through a regular elbow.

All piping between the boiler and heater, as well as between the heater and the storage tank, must be free of sediment and lime deposits. If flushing these lines out will not clear them, take them down and clean thoroughly, or replace with new pipe. I prefer to replace the piping.

Reaming the piping is very necessary, but on practically every job which we have taken down we find that not one piece has been reamed, except for the mill ends.



Improvised Stoker Drive Supplies Heat During Flood

During the recent flood when the downtown section of Pittsburgh was inundated and without electric service, heat for the buildings of almost a block was supplied from the heating plant in the Berger Building. Several small gasoline engines were collected from concrete mixers and rigged up through pulleys and belts to drive the pumps and stoker. The illustration* shows one of these engines driving the stoker unit serving a 150 hp boiler.

*Courtesy Combustion Engrg. Co., New York, N. Y.

Fig. 1 Ballroom Air Conditioning System

Design and Control Complicated by Varying Day and Night Heat Loads

By F. W. Rabe, M. E.*

LAST year, the main dining room, coffee shop, barber shop, a private dining room, and the studios of Radio Station KRLD at the Adolphus Hotel, Dallas, Texas, were air conditioned, central plant air conditioners supplied with chilled water from a refrigerating plant in the basement being utilized. This year the hotel has air conditioned its magnificent junior ballroom, Figs. 1 and 2, as well as additional private dining rooms; unusual in shape, the ballroom presented a problem in the location of outlets for correct distribution, it being necessary to install ceiling plaques supplemented by side wall grilles in some portions of the space, and in the supplying of the proper air quantities to meet load demands.

Ammonia Compressors Supply Refrigeration

At the time of the original installation last year, the hotel's refrigeration requirements were supplied by two steam driven ammonia compressors and one motor driven ammonia compressor. As two of them had been in service over 20 years and the third, while only 10 years old, was steam driven and did not fit in with the scheme of things, it was decided to scrap or sell them.

A horizontal, two cylinder, steam driven air compressor was at hand which served a number of air lifts for pumping water from deep wells until the air lifts were replaced with deep well pumps. It was decided to replace the air cylinders of this machine with ammonia cylinders; this was done, resulting in an ammonia compressor having a capacity of 155 tons, Fig. 3. In addition, one 9 in. by 9 in. and one 8 in. by 8 in. motor driven two cylinder vertical compressors were purchased. To cool the water for air conditioning, ammonia at 50 lb pressure is used; for ice making and cold storage work, ammonia at about 25 lb suction pressure is employed. Each compressor can be operated on either the high pressure or low pressure suction line, and by means of capacity and speed control the desired suction pressure for given loads can be maintained.

A second 9 in. by 9 in. ammonia compressor was put in this year to take care of the additional refrigeration required for air conditioning the junior ballroom.

*Consulting Engineer, Dallas, Texas.

Variations in design conditions for an air conditioning system complicate selection of the equipment and its control. In a hotel ballroom, used for conventions during the day and for dancing and dining at night, total heat gain as well as the percentage of latent to total heat gain changes, must be allowed for.....The consulting engineer describes here the solution of the problem for the junior ballroom at the Hotel Adolphus, Dallas.....The Adolphus, headquarters for the A. S. M. E. meeting this month, will also be the Dallas home of many who will visit the Texas centennial exposition this summer

Day and Night Ballroom Cooling Loads

In designing the ballroom system, two major conditions were considered, (1) use during the day for conventions with the load calculated upon an outside design dry bulb of 98 F and a 78 F wet bulb, 1,000 people at rest, winter garden not in use, and 3,000 watts of lighting as listed in Table 1; (2) for night use as a ball room with 1,000 people, 3,000 watts of lighting, winter garden in use, 10,000 cfm outside air at 88 F dry bulb and 75 F wet bulb, as listed in Table 2.

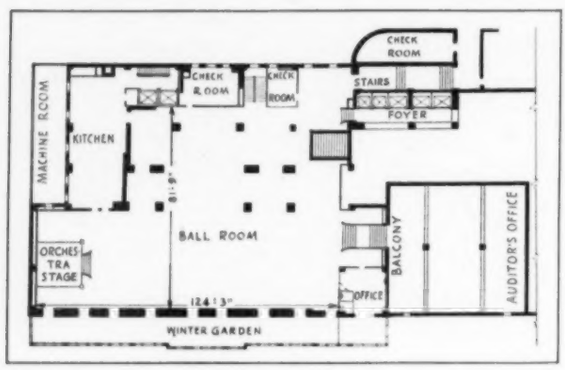


Fig. 2—Plan of the ballroom at the Adolphus

It will be noted that the percentage of latent heat to the total heat for the day load is 42.6 per cent, for night, 54.7 per cent, and that the night total load is greater than the day total load, while the day conduction load is 43 per cent greater than the night conduction load. It was necessary to design the system so as to control the design dry bulb of 78 F and 65 F wet bulb inside to hold constant during any variation due to changes from the day design to night design conditions, at the same time keeping the air volume to a minimum and the conditioning equipment of minimum size to fit the space available in the machine room.

Table 1—Day Cooling Load, Junior Ballroom, Btu per Hr

Sensible Heat	
Conduction	206,850
Outside air	177,000
People	220,000
Lighting	10,200
Total, sensible heat	614,050
Latent Heat	
Outside air	277,200
People	180,000
Total, latent heat	457,200
Total heat load	1,071,250

Table 2—Night Cooling Load, Junior Ballroom, Btu per Hr

Sensible Heat	
Conduction	144,642
Outside air	106,000
People	270,000
Lighting	10,200
Total, sensible heat	530,842
Latent Heat	
Outside air	261,000
People	380,000
Total, latent heat	641,000
Total heat load	1,171,842

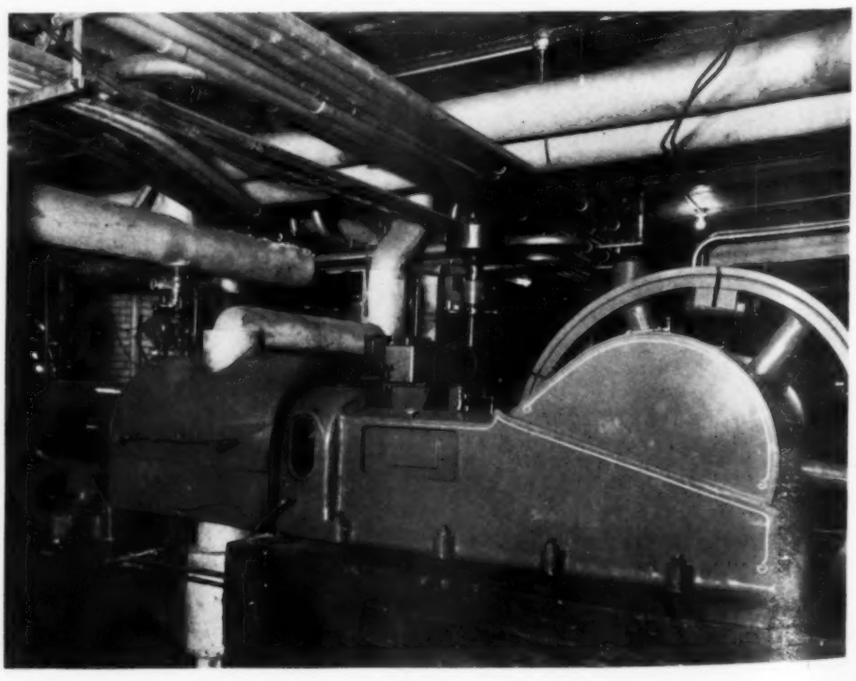


Fig. 3—Rebuilt air compressor now chilling water for air conditioning

Evaporator Design

In designing the evaporator, it was necessary to take into consideration volume of the conditioned space, method of refrigeration, available space for equipment, and condition of air entering and leaving the evaporator to maintain design conditions.

The volume of the conditioned space is 242,000 cu ft. To stay within the limits of good practice it was decided to circulate not less than 24,000 cfm in order to obtain approximately six air changes per hour. Because it was decided to use cooling water having a temperature of 42 F for the initial air conditioning installation, it was necessary to use water at the same temperature in the conditioner for the junior ballroom, as all water coolers are connected to the same ammonia system.

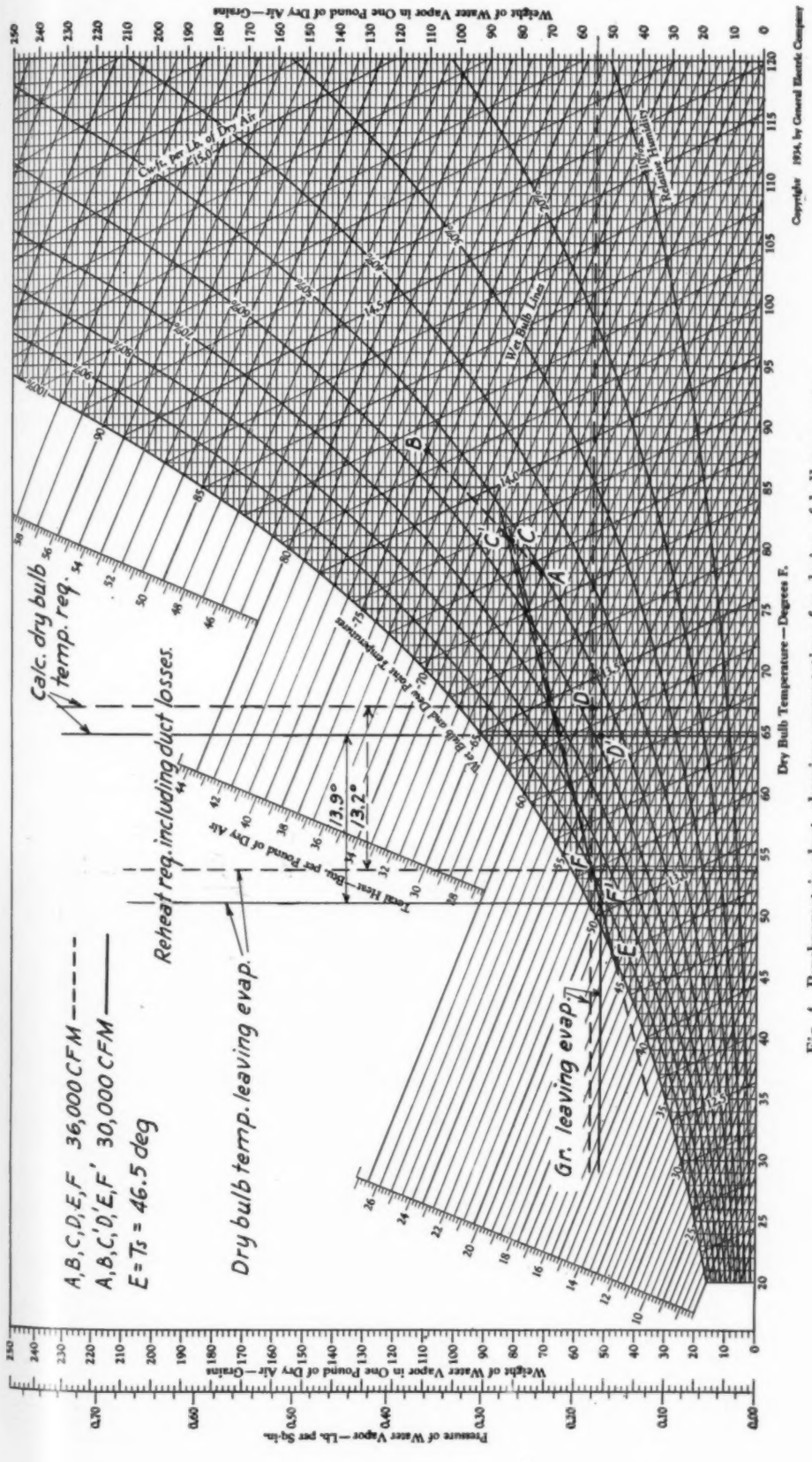

Weight of Water Vapor in One Pound of Dry Air—Grains
Dry Bulb Temperature—Degrees F.
Wet Bulb Lines
Relative Humidity Lines
Copyright 1934, by General Electric Company
Calc. dry bulb temp. req.
Reheat req. including duct losses.
Dry bulb temp. leaving evap.
Gr. leaving evap.
13.9°
13.2°
A,B,C,D,E,F 36,000 CFM
A,B,C,'D,'E,'F' 30,000 CFM
E = Ts = 46.5 deg
Pressure of Water Vapor—Lb. per Sq. In.

Fig. 4—Psychrometric chart, showing computations for design of ballroom system
A, inside design conditions. *B*, outside air. *C*, air entering evaporator.

With a fixed water temperature of 42 F, it was necessary to determine the lowest surface temperature possible in order to obtain maximum dehumidification and at the same time keep the water rate within an economical operating range and the velocity high enough to give as low a coil surface temperature as possible. This was determined from heat transfer coefficients given by the evaporator manufacturer for a face velocity of 500 fpm and various water velocities. In this case, the surface temperature T_s of 46.5 F was found to be most economical from the standpoint of pumping costs. Using an average surface temperature of 46.5 F, it was found that 36,000 cfm would meet the load demand with 80.78 F dry bulb temperature and 67.8 wet bulb temperature entering the evaporator, with a moisture drop of 26.8 grains per lb of dry air flowing across the cooling coils, and a 13.7 F temperature drop.

Plotting *A*, *B*, *C*, and *D* on Fig. 4, it was found impossible to get a cooling line through *C*, *D*; it was noted that reheat was necessary during the maximum latent heat load. Cooling line *CE* was drawn crossing the grains of moisture line at *F* which

indicated that reheat of 13.2 F was necessary to meet the design condition or an additional 42 tons of reheat, giving a total evaporator load of 139.7 tons. Since the 42 tons of reheat are necessary for night operation at a given design condition, this reheat is reduced as the sensible heat load in the conditioned area increases; without further calculation, it is seen that these coils will handle the increase in conduction load during day operation.

How do we know the air volume as calculated will enter the conditioned area as indicated at *F* on the psychrometric chart? Since this point was determined as the correct dew point on the cooling line to meet the designed condition, the proper coil was selected with a face velocity of 500 fpm. Having already determined the surface temperature T_s, dry bulb temperature and wet bulb temperature of both the air entering and leaving the evaporator, the manufacturer's heat transfer coefficients from the air to the fin surface were used to select the proper coils and size for the calculated load. Should the cfm change from 36,000 to 30,000, as might be the case because of increase in duct resistance due to extra ells not considered in selecting the fan, a reduction in reheat would occur; in this case, it would be 4.6 tons as indicated by *A, B, C', D', E, F',* allowing 4.6 tons extra capacity available to take care of errors in calculations.

Fig. 5—Machine room panel board for operation of Adolphus Hotel air conditioning system

Major equipment for the ballroom air conditioning system comprises four sections of finned coils having a total surface of 19,344 sq ft, 65 sq ft face area, made of ¾ in. tubes, each section eight coils high and four deep; 36,000 cfm fan driven through V-belts by a 15 hp motor; glass wool air filters; and the necessary thermostats, humidistats, grilles, motorized valves, etc. The eight pass shell and tube water cooler has an external surface of 1,391 sq ft, and there are two cold water pumps, one 450 gpm and the other 250 gpm.

Method of Control

Because of the variable load and necessity of reheat, a control system to control both the dry bulb and relative humidity in the conditioned area was required. This was obtained by installing in the return duct from the conditioned area a proportioning duct thermostat and proportioning duct humidistat. The duct humidistat operates a proportioning positive closing steam valve in conjunction with a duct limit control located past the reheat coil set for maximum high dry bulb temperature and corrected for duct conduction losses to prevent the reheat coil from over-shooting should the steam valve attempt to open suddenly. The duct thermostat operates a water proportioning valve which in turn operates a two position valve by means of an auxiliary switch on the shaft of the proportioning water valve. A two position water valve is installed in the water line to cut off one-third of the total cooling surface in vertical rows of the entire conditioner face.

Because the winter garden is used only at night, a proportioning duct damper operated by means of a room proportioning thermostat in the winter garden was installed in the supply duct. To close the damper during day operation, a double pole, double throw switch was installed on the panel board to add a given resistance to the damper motor and close the damper when the switch is set in the off position.

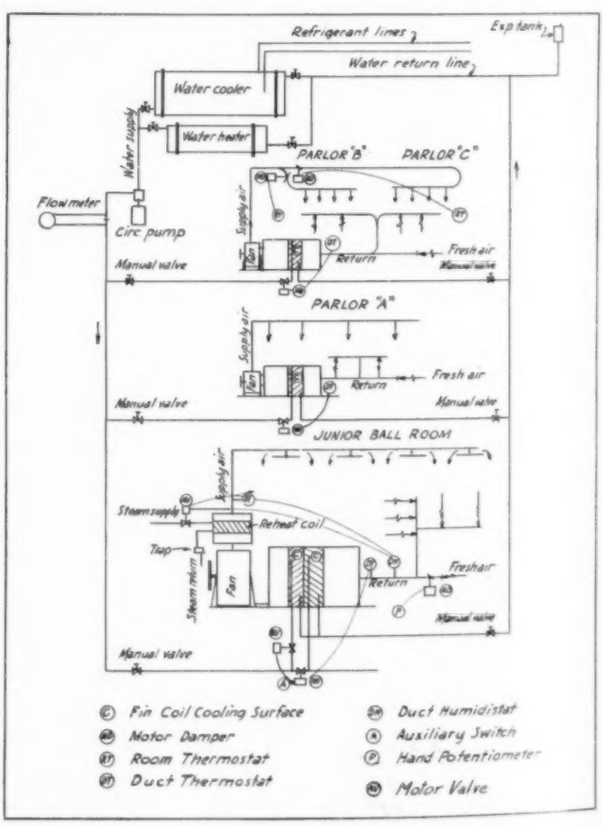


Fig. 6—Air conditioning system flow diagram

The fresh air load was determined by using 10 cfm per person or a 28 per cent dilution by fresh air. The fresh air damper is operated manually at the panel board by means of a hand potentiometer operating a proportioning damper with damper set at maximum opening for 10,000 cfm.

Reheat coils were designed to reheat the fresh and recirculated air during winter operation with a manual damper to increase capacity of coils from summer to winter operation. The control of the steam valve is accomplished by a selector switch to transfer the duct thermostat from the water valve to the steam valve and reverse the proper electrical leads for valve to operate upon a fall in temperature rather than a rise as it would in summer operation.

The panel board, Fig. 5, is in the machine room and is equipped with temperature indicating gauges for incoming water, leaving water, return air dry bulb temperature, dry bulb temperature of air leaving cooling coils, and dry bulb temperature leaving reheat coils. This panel board also includes an electrical control selector switch for two private dining rooms which operate from one conditioner with maximum capacity for both rooms with proper contacts to change speed of motor and close off duct dampers on rooms not requiring conditioned air.

This panel board is fully equipped with all necessary gauges, push button station, indicator lights and room control selector switch.

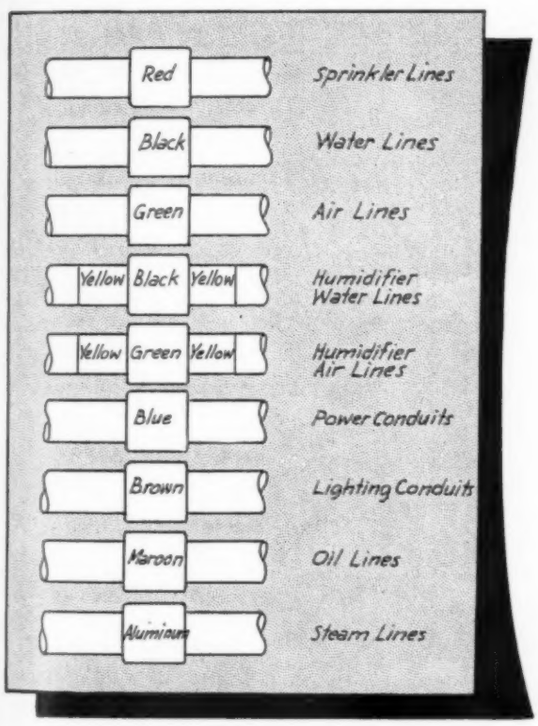
Duct Work

The duct work was installed in furred ceiling space above the ballroom. Ceiling plaques were designed to match the present ceiling and give an attractive appearance.

Fin type grilles were used for all the returns, with half the returns under the balcony 11 ft above the floor. Velocity through the grilles is 600 fpm.

The design and installation of the duct work conformed to the standards as given in the *A. S. H. V. E. Guide*. Both supply and return ducts are insulated with ½ in. board insulation sealed with a waterproof cement.

Contractors for this installation were Electric Household Appliances, Inc., and the Dallas Heating and Ventilating Co.



Plant Adopts Piping Identification Scheme

By **W. W. Watkins***

THE drawings show the scheme for plant piping identification we have recently adopted. According to the plan, all piping except valves and fittings is painted the same color as adjacent surfaces of walls or ceilings; vertical pipe lines adjacent either to walls or columns are painted the same color as the dado for the height of the dado and the rest of the line is painted the same color as the ceiling. This is modified in the case of a line rising from a machine, when the piping is painted the same color as the machine up to a height of 5 ft above the floor.

Fittings and valves are painted according to the chart, depending upon the piping service. It is not felt necessary to paint all pipe couplings and fittings where they are close together, but only enough to identify the line easily and quickly; thus these are marked at intervals of from 10 ft to 25 ft, depending upon the piping layout and how easily a line can be traced. Where pipe lines go through walls or floors, identifying bands are painted on the piping on each side of the wall or floor 1 or 2 ft from the wall surface. All valves are painted according to the color scheme shown.

(For further information on identification of plant piping systems, see "Standard Adopted for Identification of Piping Systems," HEATING, PIPING AND AIR CONDITIONING, July, 1929; also "Identify Plant Pipe Lines to Save Time and Mistakes," by Robert W. Cuthill, HEATING, PIPING AND AIR CONDITIONING, April, 1934. —THE EDITOR.)

*Tire Cord Division, United States Rubber Products, Inc., Hogansville, Ga.

Reliable Steam Consumption Records Mean Better Heating at Minimum Cost

By **Lyman R. Flook***

> ***"When you cannot measure, your knowledge is meager and unsatisfactory."—Lord Kelvin.***

This quotation is cut in stone on the Social Science Building at the University of Chicago, the first building designed and built exclusively for the study of the social sciences.

This maxim from a great scientist should be the slogan for the modern operating engineer or for the up-to-date service organization. It has been taken quite literally by our organization, and applied as our first principle in the fascinating problem of conserving the steam used in our heating service. It has really been the keynote of our effort in the management of the use of steam for heating.

In the hope that a more complete description of our present practice can be adapted or applied by others with similar operations, this practice, improved in recent years, is presented here. A previous article[1] described the organization and result of the heating steam conservation efforts at the University of Chicago over a period of years.

For efficient operation of the heating service for a building, an industrial plant, or for a group of buildings, records must be adequate, reliable and prompt. If any one of these three needs is inadequately supplied, service suffers or costs will be high.

For adequate records, meters must be reliable. To assure accuracy of readings we have a meter testing laboratory, Fig. 1, in which actual weights may be measured against meter dial readings, and the meter factor determined. This test is made on every meter at least once a year, and during the heating season if indications of trouble warrant it.

The test record and calibration factor or multiplier for each meter are entered on the back of a meter record card; on the front of this same form the monthly readings are entered in the office from the meter reader's field book, Fig. 6. The office engineer in charge of these cards subtracts meter readings, applies the factor and makes up the steam bill for each building, typing on each monthly bill the amount of steam used in 1000 lb, the rate per 1000 lb, the amount in dollars, and the Factor "X" which is calculated for each building monthly, and which is pounds of steam used that month per degree-day per 1000 cu ft of building volume.

Just as soon as the calculations for the bills are completed a steam use report is made, showing steam used for heating, for hot water, and in a few cases for cooking, laboratory or other special use, as well as the "losses," which include discrepancies in meter readings, steam not returned, and losses in transmission. A sample of the steam use report and a steam bill were shown in the November, 1935, issue.

From these records another analysis is made by buildings and by groups of buildings, for assisting in the control of steam for heating. For each month, steam used for heating, in 1000 lb, is entered on a work sheet, months vertically, on which the buildings are grouped in seven classifications. From these work sheets a single master sheet, called the steam summary is typed, Fig. 7. This master sheet is a very interesting and valuable monthly report, giving at once the variation in each classification, either plus or minus, and sub-totals for the two groups, each compared with the same month of the previous year. In addition, it lists the ten buildings showing the greatest increase of use for the month.

These monthly reports, accumulating, show the trend and enable management to determine steam conservation policies. Plotted, Fig. 8, these monthly summaries show an interesting use of the Factor "X". The widely fluctuating shapes of these curves for September, October, May and June, the months of least steam use when the degree-day total is small, do not invalidate the very consistent curves for the winter months, when the degree-day total is large—the six months of high heating expense in which management is most keenly interested.

Fig. 9 shows the steam consumption for one of the seven groups into which the buildings are divided. To

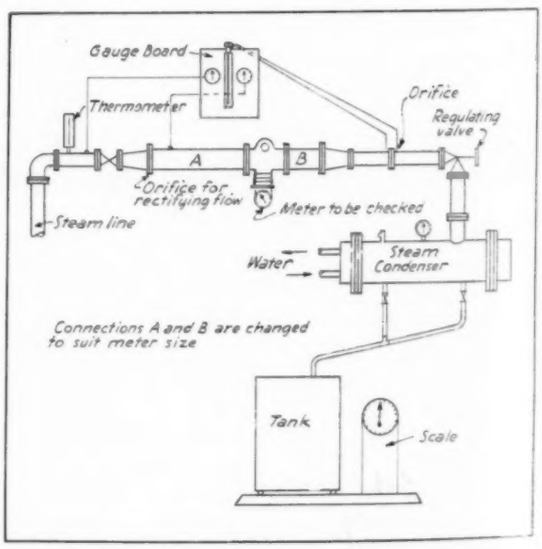
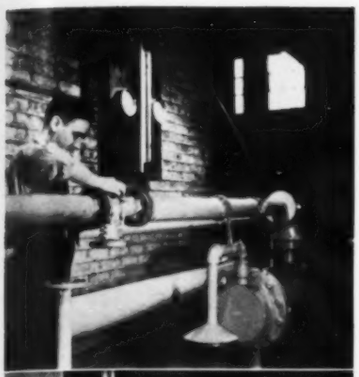


Fig. 1—Arrangement for testing steam meters

*Superintendent of Buildings and Grounds, The University of Chicago.

[1] "Persistent Conservation Efforts Cut Steam for Heating in Half," by Lyman R. Flook, HEATING, PIPING AND AIR CONDITIONING, November, 1935, pp. 513-517.



Now is the time to analyze heating plant performance for better, more economical operation next winter, a task that is simplified if adequate records in the proper form are available as they are at The University of Chicago, whose methods are described here

THE UNIVERSITY OF CHICAGO
The Department of Buildings and Grounds

April 2, 1936

STEAM SUMMARY

March - 1936

	M-lbs.			Factor "X"		
	1934-35	1935-36	% Diff.	1934-35	1935-36	% Diff.
Group						
(A) B and G General*						
I Academic	7,426	8,046	+ 8.3	.647	.608	- 6.0
IV Laboratory	5,043	5,566	+10.4	1.108	1.060	- 4.3
V Athletic	2,631	3,336	+26.8	.491	.540	+10.0
VI Administr.	891	1,183	+32.8	.613	.706	+15.2
Total	15,991	18,131	+13.4	.700	.688	- 1.7
(B) Others *						
II Dormitories	2,979	3,422	+14.9	.870	.866	- .5
III S.S.Med.Grp.	5,808	5,976	+ 2.9	1.110	.990	-10.8
VII Affiliated	5,842	6,480	+10.9	1.440	1.358	- 5.7
Total	14,629	15,878	+ 8.5	1.151	1.076	- 6.5
GRAND TOTAL	30,620	34,009	+11.1	.861	.828	- 3.8

LIST OF TEN HIGHEST INCREASES OVER LAST MARCH

(A) Group I	Bond Chapel	.736	.820	+11.4
	Classics	.469	.595	+26.9
	Wieboldt	.433	.523	+20.8
Group IV	Capon House	2.052	2.833	+38.1
	Culver	1.254	1.454	+15.9
	Whitman Lab.	1.462	1.710	+17.0
	" An. Bldgs.	2.718	3.702	+36.2
Group V	West Stand	.748	1.112	+48.7
Group VI	Storeroom	.917	1.012	+10.4
Group VII	Quadrangle Club	.963	1.158	+20.2

* On a cubage basis (A) is 96% B and G General and 4% Others. (B) is 99.5% Others and .5% B and G General.

Fig. 7—Steam summary for March, 1936

separate the steam used by each group, sub-master or branch meters are needed, and to allocate the several "losses" these group sub-meters are compared with the meters in the individual buildings of the group. By plotting the group sub-meter readings in this way each on a separate graph, the group steam use and losses are clearly and simply shown.

Fig. 10 is a master record and budget forecast for the system as a whole. The use of some such means of predicting the cost of heat and comparing actual cost with the predicted cost each month is strongly recommended. Curve *A* is the forecast, based upon the average of the two preceding years adjusted on a degree-day and probable use basis, the budget estimate; Curve *C* is last year's actual cost; Curve *D* the second preceding year's actual cost; and Curve *B* the cost for the present year, extended monthly.

The upper curves of Fig. 10 are by months and the lower are cumulative.

Some curiosity has been expressed with regard to quantitative results of the past year. For purposes of further illustration the result of the operations for the year 1934-35 are shown in summary by the graphs of Fig. 11. These graphs show the University buildings by groups. In each group the unit use of steam, expressed as Factor "X," varies widely by reason of variations in use, construction,

TOP TO BOTTOM

Fig. 2—Installing body of shunt meter preparatory to testing. . . . Fig. 3—Meter testing engineer calibrating shunt type meter. . . Fig. 4—Reading condensation type building meter. . . . Fig. 5—Reading orifice type building flow meter. . . . Fig. 6—Office engineer entering readings on record cards

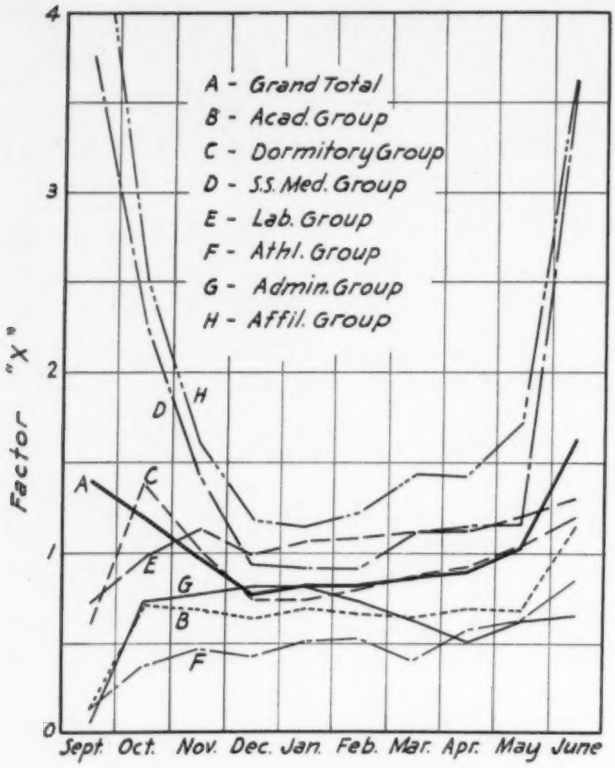

A - Grand Total
B - Acad. Group
C - Dormitory Group
D - S.S. Med. Group
E - Lab. Group
F - Athl. Group
G - Admin. Group
H - Affil. Group
Factor "X"
Sept. Oct. Nov. Dec. Jan. Feb. Mar. Apr. May June

Fig. 8—Pounds of steam per degree-day per 1000 cu ft of building volume (Factor "X") for the 1934-35 heating season

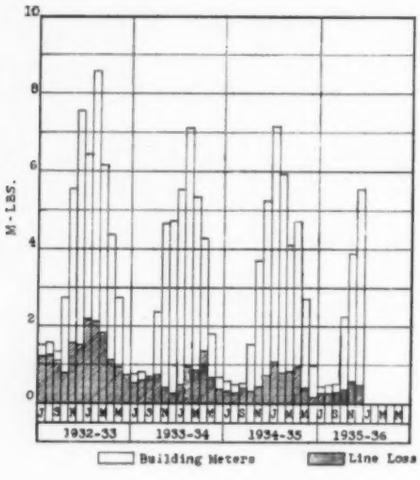
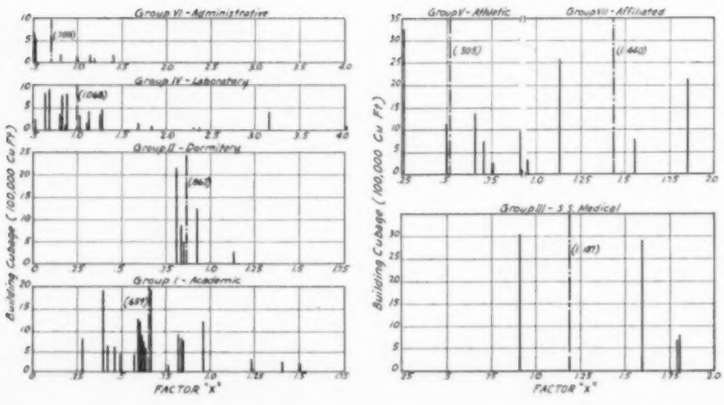


Fig. 9—Steam consumption and line loss for one of the groups


Group VI - Administrative
Group IV - Laboratory
Group II - Dormitory
Group I - Academic
Group V - Athletic
Group VIII - Affiliated
Group III - S.S. Medical
Building Cubage (100,000 Cu Ft)
FACTOR "X"

and design. The extent of this variation is shown for each group by the vertical lines; each vertical line represents a separate building. The length or height of each line gives the size of the building (in hundreds of thousand cubic feet), and by its location on the base line the value of Factor "X." The broken line in each group is the average unit steam consumption, and the figure in

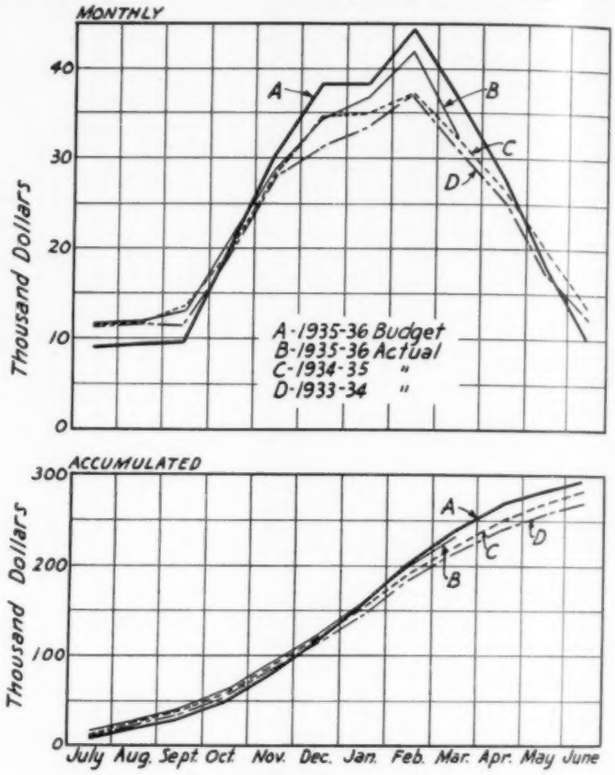


Fig. 10—Master record and budget forecast. *A*, forecast based on average of two preceding years; *B*, present year, extended monthly; *C*, actual for preceding year; *D*, actual for second preceding year. Upper curves by months, lower, accumulated

parenthesis is the average for the group in lb per degree-day per 1000 cu ft of building volume.

The larger buildings are generally to the left, usually average or better, representing in general the more recent construction with better design and control, while the smaller and older ones are more generally to the right, or in the region of greater steam use, being for the most part the older and smaller buildings, except for one extreme right line representing a very large greenhouse which must be heated continuously.

Our methods of assuring accurate meter readings and the use of these data have been described in some detail, as the procedure is applicable not only to institutional buildings but to plant and commercial buildings as well. Now is the time when heating plant performance during the past winter should be given consideration in order that maintenance and modernization during the summer can be planned and the system put in shape for better results next year. This is simplified if performance records are available in a form permitting of analysis.

Fig. 11—Annual use, by groups, with mean for each group

Solving the Four Types of Process Air Conditioning Problems

By **William Goodman***

All air conditioning problems fall into one of four main divisions, depending upon whether the sensible and latent heat are being subtracted from, or added to, the space being conditioned. Problems typical of each division are discussed here and solutions by the psychrometric chart explained

[Part 1]

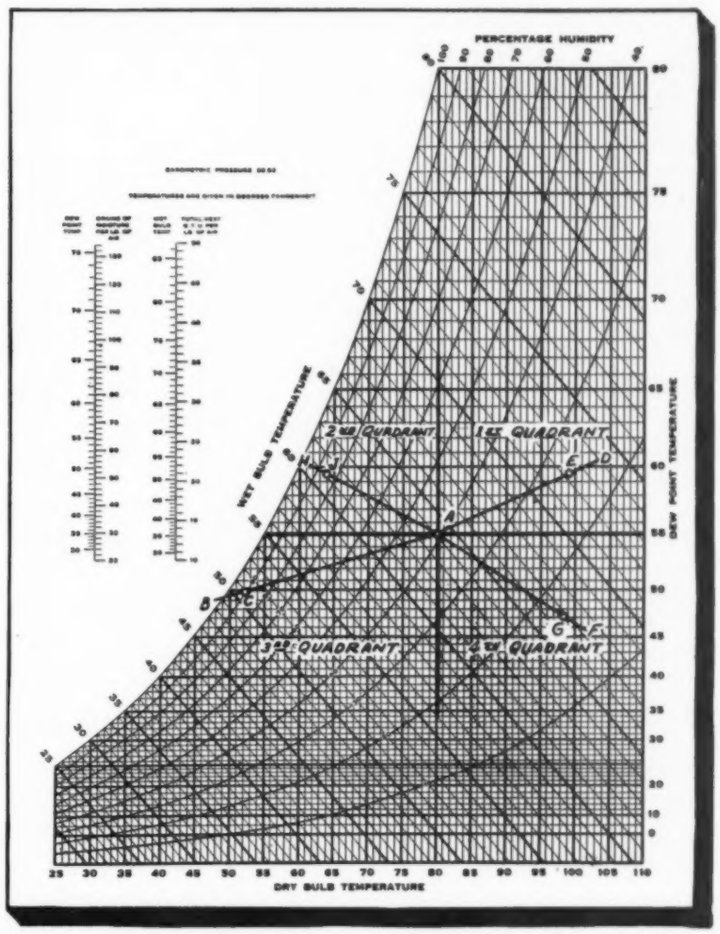


Fig. 1—The point representing the condition of the air supply must lie in the third quadrant for cooling and dehumidifying, the first quadrant for heating and humidifying, the fourth quadrant for heating and dehumidifying, and the second quadrant for humidifying and cooling. See Table 1 on the next page

ALL air conditioning problems, whether comfort or industrial process, must fall into one of four main divisions which are fundamental and depend upon whether the sensible and latent heat are being subtracted from, or added to, the space being conditioned. In summer comfort cooling, both sensible and latent heat are being subtracted from the conditioned space; therefore, the point representing the condition of the air supply must lie in the third quadrant of Fig. 1 if the room condition represented by the point *A* is to be maintained. As the air, initially in the condition represented by the point *C*, heats along a "heat ratio line" such as *A-B* it absorbs both sensible heat and moisture in the conditioned space.[1] On the other hand if, as is often the case, a room must be heated and humidified to offset the loss of sensible heat and moisture, then the point representing the condition of the air supply must lie in the first quadrant of Fig. 1. As the air initially in the condition represented by the point *E* cools along a heat ratio line such as *A-D*, it surrenders both sensible heat and moisture to the conditioned room.

There are also processes during which a conditioned space may be losing sensible heat and at the same time gaining moisture. This is true of practically all drying processes, especially during the winter months. In this case the air supply surrenders sufficient sensible heat to offset the loss of such heat from the room and at the same time absorbs moisture in the conditioned space. To accomplish this, the point representing the initial condition of the air supply must lie in the fourth quadrant of Fig. 1. As the air supply initially in the condition represented by the point *G* cools along the ratio line *A-F* it surrenders sufficient sensible heat to offset the loss of this heat from the room. However, while cooling, the dew point temperature of the air supply is increasing, with the result that moisture is absorbed.

The last of the divisions into which an air conditioning process may fall is one in which humidification is required while the sensible heat gain must be absorbed. In order to satisfy the requirements of such a process, the point representing the initial condition of the air

*The Trane Co., LaCrosse, Wis. Copyright, 1936, by William Goodman.

[1]For a discussion of the heat ratio line, see HEATING, PIPING AND AIR CONDITIONING, February, 1936, p. 90.

supply must lie in the second quadrant of Fig. 1. As the air supply in the initial condition represented by the point *J* heats along the ratio line *A-H*, it absorbs sensible heat in the conditioned space. At the same time, since the dew point temperature of the air supply is falling, moisture is being supplied to the conditioned space to offset the loss of moisture which is taking place.

Table 1 will help in clearly fixing in mind these four main divisions.

Heating and Humidifying

Problems in which both sensible heat and moisture are being lost from the conditioned space are handled in the same manner as those in which they are being gained, except that the initial condition of the air supply, as can be seen in Table 1, must lie in the first quadrant instead of the third. The heat ratio angle is always measured from the horizontal line, as shown in Fig. 2, where the ratio line *A-B* has been drawn for the conditions of Example 1 below. Heat ratio angles are always measured from the horizontal line through the point representing the room condition, regardless of whether the ratio line lies above or below the horizontal line through the room condition point.

EXAMPLE 1—A room whose dry bulb temperature is to be maintained at 68 F has a sensible heat loss of 74,000 Btu per hr. In addition it is estimated that 19 lb per hr of water vapor will be required to maintain the relative humidity at 45 per cent. If the dry bulb temperature of the air supply is to be 95 deg, find the required wet bulb temperature of the air supply.

Solution—

Latent heat of vapor = 1050 Btu per lb

Latent heat loss = $19 \times 1050 = 19{,}950$ Btu per hr

Referring to Table 1, both the sensible and latent heat losses are to be considered as negative.

$$\text{Sensible heat ratio} = \frac{-74{,}000}{-74{,}000 - 19{,}950} = \frac{-74{,}000}{-93{,}950} = 0.787$$

[When dividing one negative number by another negative number, the answer is always a positive number. When dividing a negative by a positive number, or *vice versa*, the answer is always a negative number.]

Referring to Table 2.

Heat ratio angle = 18° —42′ (by interpolation)

The point *A* of Fig. 2 represents the required room condition of 68 deg dry bulb and 45 per cent relative humidity.

Referring to Table 1, the heat ratio line must lie in the first quadrant. Draw the heat ratio line *A-B* in Fig. 2. Spot the point *C* where the vertical line of 95 F dry bulb temperature crosses the ratio line *A-B*. The required wet bulb temperature of the air supply, as read at the point *C*, is 67.9 F.

Cooling and Humidifying

Spaces used for process work whose humidity must be maintained at a high point while their dry bulb temperature must be kept low produce the odd situation of a room which must be cooled and at the same time humidified. When this is the case, the initial condition of the air supply must lie somewhere in the second quadrant. Problems involving the second quadrant divide into two classes, depending upon whether the sensible heat gain is greater or less than the loss of latent heat.

Table 1

Conditioned Space	Air Supply	Quadrant in Which Heat Ratio Line Must Lie	Sign to be Used in Calculating Sensible Heat Ratio
Loses sensible heat Loses moisture	Loses sensible heat Loses moisture	1	− −
Gains sensible heat Loses moisture	Gains sensible heat Loses moisture	2	+ −
Gains sensible heat Gains moisture	Gains sensible heat Gains moisture	3	+ +
Loses sensible heat Gains moisture	Loses sensible heat Gains moisture	4	− +

Table 2—Heat Ratio Angles for Various Sensible Heat Ratios, 1st and 3rd Quadrants

Heat Ratio	Heat Ratio Angle	Heat Ratio	Heat Ratio Angle
0.00	90°-0′	0.50	51°-20′
0.02	89°-4′	0.52	49°-5′
0.04	88°-5′	0.54	46°-48′
0.06	87°-5′	0.56	44°-29′
0.08	86°-1′	0.58	42°-9′
0.10	84°-55′	0.60	39°-48′
0.12	83°-46′	0.62	37°-27′
0.14	82°-35′	0.64	35°-7′
0.16	81°-20′	0.66	32°-47′
0.18	80°-2′	0.68	30°-28′
0.20	78°-42′	0.70	28°-11′
0.22	77°-17′	0.72	25°-56′
0.24	75°-49′	0.74	23°-42′
0.26	74°-18′	0.76	21°-33′
0.28	72°-43′	0.78	19°-25′
0.30	71°-4′	0.80	17°-21′
0.32	69°-22′	0.82	15°-21′
0.34	67°-36′	0.84	13°-24′
0.36	65°-46′	0.86	11°-30′
0.38	63°-53′	0.88	9°-41′
0.40	61°-56′	0.90	7°-54′
0.42	59°-55′	0.92	6°-12′
0.44	57°-51′	0.94	4°-34′
0.46	55°-41′	0.96	2°-59′
0.48	53°-33′	0.98	1°-28′
		1.00	0°-0′

Table 3—Heat Ratio Angles for Various Sensible Heat Ratios, 2nd and 4th Quadrants

Heat Ratio	Heat Ratio Angle	Heat Ratio	Heat Ratio Angle
1.00	0°-0′	−3.00	59°-3′
1.10	6°-29′	−2.80	59°-29′
1.20	11°-46′	−2.60	59°-59′
1.30	16°-6′	−2.40	60°-33′
1.40	19°-39′	−2.20	61°-12′
1.50	22°-37′	−2.00	61°-56′
1.60	25°-7′	−1.90	62°-21′
1.70	27°-14′	−1.80	62°-48′
1.80	29°-3′	−1.70	63°-16′
1.90	30°-38′	−1.60	63°-47′
2.00	32°-0′	−1.50	64°-21′
2.20	34°-17′	−1.40	64°-59′
2.40	36°-6′	−1.30	65°-40′
2.60	37°-34′	−1.20	66°-26′
2.80	38°-47′	−1.10	67°-16′
3.00	39°-48′	−1.00	68°-12′
3.50	41°-46′	−0.90	69°-15′
4.00	43°-9′	−0.85	69°-49′
5.00	44°-60′	−0.80	70°-26′
6.00	46°-10′	−0.75	71°-5′
8.00	47°-34′	−0.70	71°-46′
10.00	48°-22′	−0.65	72°-30′
15.00	49°-24′	−0.60	73°-18′
20.00	49°-55′	−0.55	74°-9′
30.00	50°-23′	−0.50	75°-4′
$H_S = H_L$*	51°-20′	−0.45	76°-3′
−30.00	52°-16′	−0.40	77°-7′
−20.00	52°-43′	−0.35	78°-17′
−15.00	53°-8′	−0.30	79°-32′
−10.00	53°-59′	−0.25	80°-55′
−8.00	54°-35′	−0.20	82°-24′
−6.00	55°-34′	−0.15	84°-3′
−5.00	56°-19′	−0.10	85°-50′
−4.00	57°-24′	−0.05	87°-49′
−3.50	58°-7′	−0.00	90°-0′

*When sensible heat equals latent heat, the heat ratio line coincides with the line of room wet bulb temperature.

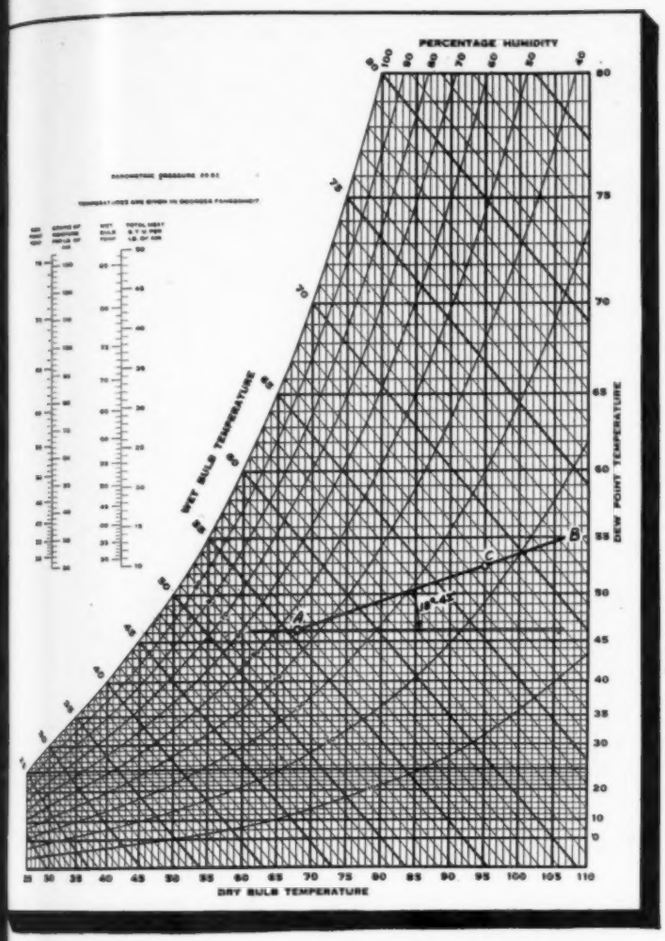


Fig. 2—Solution of Example 1, heating and humidifying

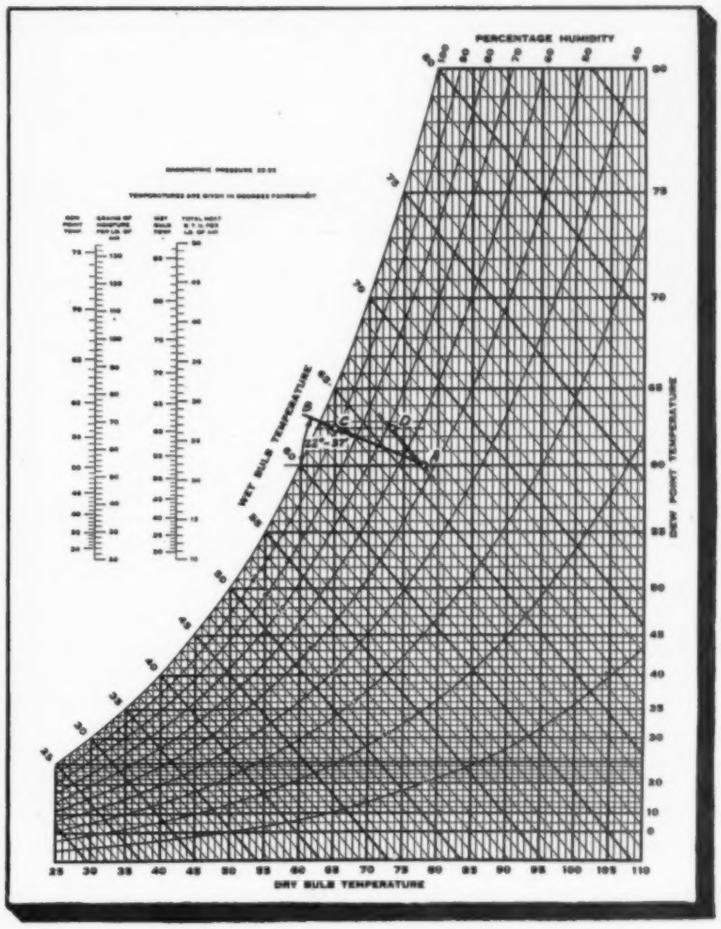


Fig. 3—Solution of Example 2, cooling and humidifying

EXAMPLE 2—A conditioned space whose dry bulb temperature is to be maintained at 78 F requires the evaporation of 25 lb per hr of moisture in order to maintain its dew point temperature at 60 F. The sensible heat gain is 79,000 Btu per hr. Find the required wet bulb temperature of the air supply if its dry bulb temperature is to be 65 F.

Solution—

Latent heat loss from room = $25 \times 1050 = 26{,}300$ Btu per hr.
(*Slide rule accuracy*)

Referring to Table 1, a loss of latent heat is to be preceded by a negative sign while a gain of sensible heat is considered as positive. Therefore,

$$\text{Sensible heat ratio} = \frac{+79{,}000}{+79{,}000 - 26{,}300} = \frac{79{,}000}{52{,}700} = 1.50$$

Referring to Table 3,

Heat ratio angle = 22° —37′

The point *A* of Fig. 3 represents the required room condition of 78 F dry bulb and 60 F dew point. The ratio line *A-B* is laid off at an angle of 22°-37′ with the horizontal line through the point *A*. The required wet bulb temperature of the air supply, as read at the point *C*, is 63.5 F.

(*To be concluded*)

Drying with Lithium Chloride

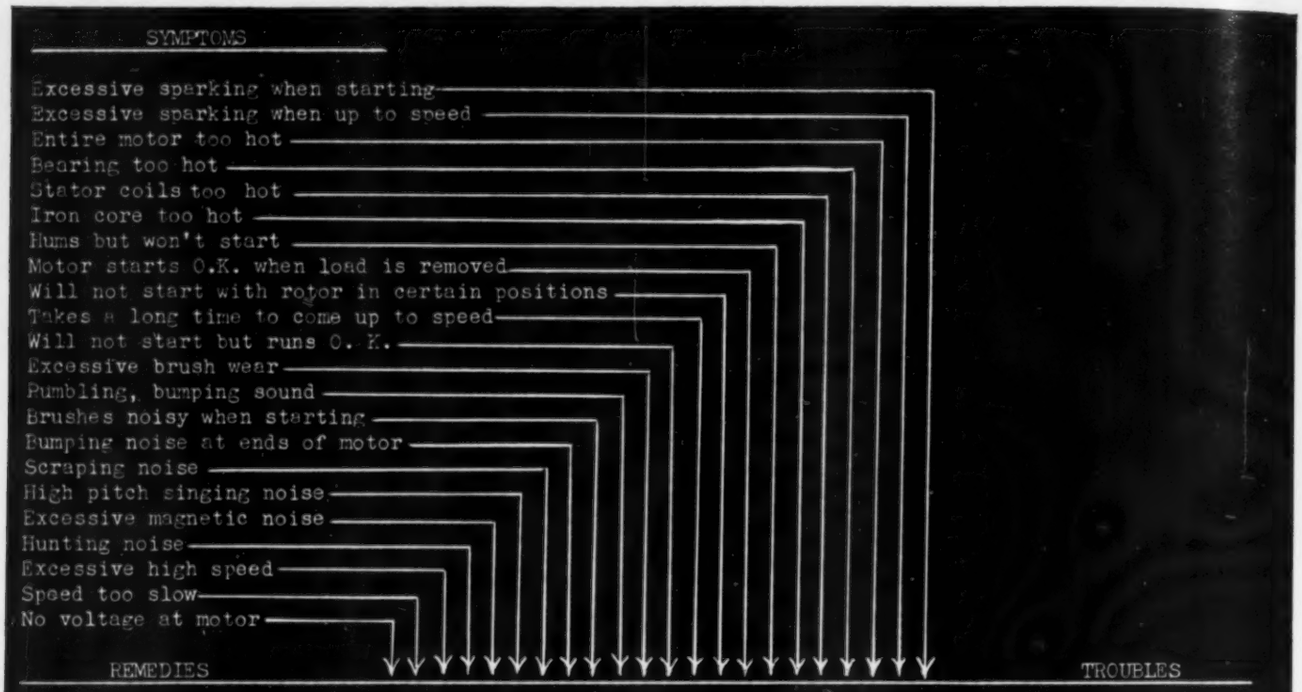
A new process of drying industrial products has been developed in which no heat need be supplied within the dryer, no fresh air is used, and no wet air thrown away. Since in ordinary air drying the heat in the waste humid air represents the major loss, the process makes possible high drying efficiencies, says the *Industrial Bulletin* of Arthur D. Little, Inc., chemists and engineers. It depends on the moisture absorbing capacity of a solution of lithium chloride.

Moist air leaving the dryer is passed in contact with a hot strong solution of lithium chloride, which absorbs the water vapor simultaneously, and heats the air. The dried hot air is then returned to the dryer, and no heat need be supplied the dryer or preheater, except to make up for conduction and leakage through the dryer walls. This is not perpetual motion, since heat must be supplied to evaporate the water from the lithium chloride solution. To boil this solution and bring it back to its original concentration requires heat, but the amount of heat needed is only slightly greater than the theoretical heat requirement for vaporizing water from the original wet material being dried. Since no outside air is used, the operation of the dryer is not subject to the vagaries of the weather.

Trouble Chart — Repulsion Induction Motors

This chart, prepared by F. W. Butler of Greer College, Chicago, for the use of students of air conditioning and refrigeration, shows the symptoms and some of the most common causes of single phase repulsion induction motor troubles, with practical remedies and tests. Often a combination of a number of symptoms makes it easier to locate the exact trouble quickly.

Code—L—Brush lifting types, R—Brush riding types, X—Both types, W—With load.



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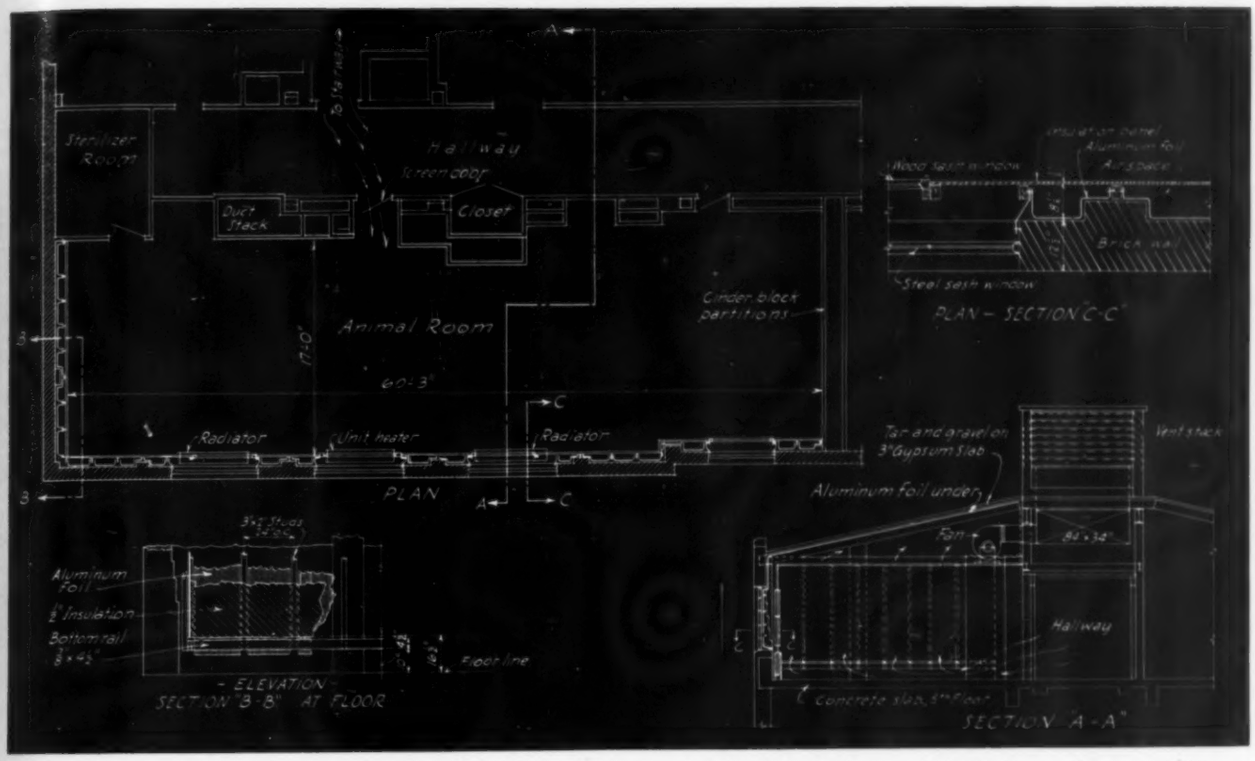

Comfortable conditions in animal room, maintained by ample fan capacity, double windows and insulation, reduce mortality rate

For Comfort, Laboratory Insulates and Ventilates

By **G. K. Saurwein***

IN many instances it is possible to provide the right atmospheric conditions for comfort economically through the use of insulation and greater air movement instead of mechanical refrigeration. Because refrigeration is usually required in comfort cooling work, so much emphasis has been placed on this phase of the subject that other means of achieving the desired result are frequently neglected. An example of the use of insulation and ventilation is therefore illustrated here.

Lack of other space in a large biology laboratory building made necessary the use of part of the attic as an animal room in which to house experimental mice, rats, guinea pigs, rabbits, cats, etc. The drawing shows the arrangement of the room, which is at the southwestern end of the building, where the sun effect is a maximum. The curtain walls are of cinder block with brick veneer, and the roof is pre-cast gypsum blocks supported on structural steel and overlaid with standard tar and gravel roofing material.

When the space was first fitted out as an animal room the contractor installed a plaster board ceiling and two small fans, one at each end of the room, to provide five air changes per hour. These provisions proved entirely inadequate for comfort; on nearly every sunny day the temperature of the room was excessive, with a resulting high mortality of the animals.

To remedy conditions double windows were installed, the outside walls and the underside of the roof were covered with aluminum foil and a false wall of plaster board was put in extending to within 10 in. of the floor, as shown in Section *C-C*. A 4,500 cfm, ½ hp fan with three speed control was installed in the attic; this fan draws air through the northerly stairwell from the lower corridors and the comparatively cool courtyard. The air flows across the room, up through the false wall and ceiling, and is discharged by the fan into the stack.

The aluminum foil and double windows reduce the heat leakage into the room to one-third the former amount and the high air flow through the wall and ceiling spaces serves to carry off the heat leakage. The increased air motion through the room improved greatly the comfort conditions and lowered the odors from the cages to an unobjectionable level.

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Table of Properties Aids Piping Design

THERE appeared with the article on support of industrial piping by F. L. Snyder* and T. E. Bridge†, published in the February, 1936, issue of HEATING, PIPING AND AIR CONDITIONING, a table of the properties of pipe. Because lack of space made it necessary to publish the table in a size too small for convenient use, and because many requests have been received for publication of these data in larger size, the information is again published. The values shown have been carefully checked by the authors, as a few typographical errors were discovered, and the corrections have been made. Data on additional pipe sizes, including thin wall piping, have been included.

It is believed that this is the most complete table available on the physical characteristics of various pipe sizes, as it usually has been necessary to consult several different handbooks and to calculate many of the values. It has been found by the authors to be a considerable time-saver in industrial piping design work.

A similar table covering commercial pipe sizes from ⅛ to 12 in. appeared last month.

Formulas for values other than shown in the table:

$$I = 0.0491\ (D_1^4 - D_2^4) = AR^2$$
$$P = 2f\ (t/D_1)$$
$$Z = 0.0982\ (D_1^4 - D_2^4)/D_1$$
$$R = 0.25\ \sqrt[2]{D_1^2 + D_2^2}$$
$$Q = \text{Ft lb}/1000 = 2.25I/y = 4.5I/D_1$$

where

P = Ultimate bursting pressure, lb per sq in.
f = Fiber stress, lb per sq in.
A = Area of metal, sq in.
I = Moment of inertia of pipe section, inches fourth
Z = Section modulus, inches cube
R = Radius of gyration.

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Q = Strength factor, representing the strength of the pipe to resist bending action. It is the resisting moment in thousands of foot pounds, the stress in the material being 27,000 lb per sq in. For other allowable stress in the material the resisting moment is directly proportional.
y = Distance of farthest fiber from neutral axis = $D_1/2$.
D_1 = External diameter of pipe, in.
D_2 = Internal diameter of pipe, in.
t = Thickness of pipe walls, in.

Pipe Wall Thickness:

Minimum thickness of pipe wall at different pressures and for temperatures not exceeding 750 F for steel or wrought iron pipe; 406 F for brass and copper pipe; and 450 F for cast iron may be determined with Formula "I" of the *A.S.M.E.* Pressure Piping Code, in which

$$t_m = (PD/2S) + C \qquad P = (2S/D)\ (t_m - C)$$

where

t_m = Minimum pipe wall thickness, in.
P = Internal pressure, lb per sq in. gauge.
(Plus water hammer allowance in case of cast iron pipe conveying liquids.)
D = Actual outside diameter of pipe, in.
S = Allowable stress in material, lb per sq in.
C = Factor for allowance for threading, mechanical strength and/or corrosion.

C VALUES

Material	*C Factor*
Cast iron pipe	0.14
Cast iron pipe pit cast	0.18
Threaded steel or wrought iron	Depth of thread
Grooved steel or wrought iron	Depth of thread
Plain end steel or wrought iron 1 in. and smaller. (Plain end includes pipe joined by flared compression couplings, laps, or van stone joints, and by welding)	0.05
Plain end steel or wrought iron sizes above 1 in. (Plain end includes pipe joined by flared compression couplings, laps, or van stone joints, and by welding)	0.065

S VALUES

Material	*Values of S*
Steel pipe seamless Grade "C" (*A.S.T.M.* A-106)	14000
Steel pipe seamless Grade "B" (*A.S.T.M.* A-106 & A-53)	11500
Steel pipe seamless Grade "A" (*A.S.T.M.* A-106 & A-53)	9000
Steel pipe lap-welded (*A.S.T.M.* A-53)	7000
Wrought iron pipe lap-welded (*A.S.T.M.* A-72)	5300
Wrought iron pipe butt-welded (*A.S.T.M.* A-72)	4500
Brass pipe seamless (*A.S.T.M.* B-43)	4500
Copper pipe seamless (*A.S.T.M.* B-42)	4000
Copper tubing seamless (*A.S.T.M.* B-75 & B-88)	4000
Cast iron pipe (*F.S.B.W.W.* P-421)	6000
Cast iron pipe pit cast (*A.W.W.A.*)	4000

[*Continued on next page*]

Power Requirements for Air Conditioning

THE amount of power required for air conditioning is dependent upon the type of application. This cannot be estimated on a cubic foot or square foot basis. A restaurant with 5000 sq ft of floor space may have many times the refrigeration load of an office building of the same space because of differences in load characteristics.

The table shows a comparison of the refrigeration tonnage of a typical department store, theater and office building, on the basis of cubical content, as well as sq ft of floor space, and indicates that the tonnage requirements and power consumption will vary with the type of load, so the calculation must be made for each job in order to determine what power consumption will be.

Type of Job	Floor Area Sq Ft	Volume Cu Ft	Tons 1000 Sq Ft	Tons 1000 Cu Ft
Dept. store	119,510	1,695,000	3.0	0.21
Office bldg.	60,730	662,540	2.8	0.26
Restaurant	9,970	117,300	9.5	0.81
Theater	16,300	635,000	8.8	0.23

From a paper by W. E. Zieber and S. F. Nicoll of York Ice Machinery Corp., York, Pa., presented at the Midwest Power Engineering Conference, Chicago, Ill., April, 1936.

The power consumption per season is becoming an important consideration and depends upon the inside and outside conditions as well as the load factors on which the installation is based. An adoption of standard recommended conditions would assist greatly both in assuring the industry against setback due to adverse public reaction to poor installations, and in allowing a more intelligent analysis of power consumption data.

The horsepower of the various motors on a given class of installation will vary with the selection of evaporating temperature in the cooling equipment, the quantity of water in circulation, the amount of air handled, and the layout of the plant. The air quantity varies with the use of from 12 to 25 F difference between the room and entering air temperature during summer operation.

The number of kilowatt hours per season per compressor horsepower is about 500 for a theater, 800 for restaurants; 100 for an office building, 900 for residences; 1100 for department stores.

Table of Properties of Pipe—continued

Properties of Pipe, 14 in. to 36 in. O. D.

Nom. Pipe Size	Outside Diam.	Inside Diam.	Wall Thickness	Wt. per Foot	Wt. of Water per Ft. Pipe	Sq.Ft. Outside Surface per Ft.	Sq.Ft. Inside Surface per Ft.	Transverse Area Sq.In.	Area Metal Sq.In.	Moment of Inertia	Section Modulus I/y	Radius Gyration Squared	Radius Gyration	ULTIMATE BURSTING PRESSURE* Butt-weld	Lap-weld	Hammer Weld	Seamless & Line Weld	Strength Factor Q 1000 Ft.Lb.
14	14	13.624	.188	27.7	63.4	3.67	3.57	146.00	8.17	203.00	29.0	24.9	4.88	-	-	-	1610	65.2
"	"	13.56	.220	32.40	63.00	"	3.55	145.00	9.53	237.00	33.85	24.8	4.88	-	-	-	1885	76.2
"	"	13.524	.238	35.00	62.50	"	3.545	144.00	10.3	244.00	34.8	23.7	4.87	-	-	-	2040	78.4
"	"	13.50	.250	36.8	62.1	"	3.54	143.00	10.82	256.00	36.6	23.6	4.86	-	-	-	2140	82.4
"	"	13.375	.3125	45.682	60.82	"	3.502	140.5	13.44	314.9	44.98	23.43	4.841	-	2232	-	2680	101.2
"	"	13.25	.375	54.568	59.68	"	3.469	137.886	16.06	372.80	53.25	23.22	4.819	-	2679	-	3210	119.8
"	"	13.188	.406	58.3	59.5	"	3.455	137.00	17.34	401.00	57.4	23.1	4.81	-	-	-	3480	129.0
"	"	13.125	.4375	63.371	58.53	"	3.436	135.3	18.64	429.1	61.29	23.02	4.798	-	3125	-	3750	137.9
"	"	13.062	.469	67.9	58.1	"	3.421	134.00	19.95	457.00	65.2	22.9	4.785	-	-	-	4010	147.0
"	"	13.00	.500	72.09	57.45	"	3.403	132.73	21.210	483.8	69.11	22.81	4.776	-	3571	-	4280	155.5
"	"	12.75	.625	89.279	55.27	"	3.338	127.68	26.26	588.5	84.08	22.41	4.734	-	4464	-	5350	189.2
"	"	12.688	.65625	93.524	54.71	"	3.322	126.4	27.51	614.2	87.74	22.33	4.724	-	4688	-	5610	197.4
"	"	12.125	.9375	130.79	49.99	"	3.174	115.5	38.47	820.5	117.21	21.33	4.63	-	6696	-	8030	263.7
"	"	11.313	1.34375	181.64	43.50	"	2.962	100.5	53.42	1082.0	154.57	20.26	4.5	-	9598	-	11500	347.8
16	16	15.624	.188	31.58	83.30	4.20	4.09	192.0	9.38	294.0	36.75	31.3	5.59	-	-	-	1410	82.7
"	"	15.524	.238	40.1	82.5	"	4.065	190.0	11.8	367.0	45.9	31.1	5.58	-	-	-	1785	103.2
"	"	15.50	.250	42.1	82.1	"	4.06	189.0	12.40	385.0	48.1	31.0	5.57	-	-	-	1875	108.2
"	"	15.438	.281	47.2	81.2	"	4.04	187.0	13.9	430.0	53.8	30.9	5.56	-	-	-	2110	121.0
"	"	15.375	.3125	52.357	80.78	"	4.025	185.62	15.40	474.0	59.25	30.77	5.547	-	1953	-	2340	133.3
"	"	15.312	.344	57.5	80.0	"	4.01	184.1	16.94	519.0	64.9	30.6	5.54	-	-	-	2580	146.0
"	"	15.25	.375	62.59	79.06	"	3.992	182.65	18.41	562.1	70.26	30.54	5.526	-	2344	-	2810	158.1
"	"	15.188	.406	67.7	78.6	"	3.98	181.0	19.89	605.0	75.6	30.4	5.515	-	-	-	3040	170.0
"	"	15.124	.438	72.7	78.2	"	3.96	180.0	21.42	650.0	81.2	30.3	5.505	-	-	-	3280	183.0
"	"	15.062	.469	77.8	77.5	"	3.945	178.5	22.9	694.0	86.8	30.2	5.50	-	-	-	3520	195.0
"	"	15.00	.500	82.77	76.49	"	3.927	176.7	24.35	731.9	91.49	30.06	5.483	-	3125	-	3750	205.9
"	"	14.938	.5313	87.77	75.83	"	3.911	175.2	25.82	774.8	96.85	30.01	5.472	-	3320	-	3980	217.9
"	"	14.625	.6875	112.43	72.73	"	3.829	168.02	33.07	971.3	121.4	29.37	5.419	-	4290	-	5150	273.2
"	"	14.5	.75	122.15	71.48	"	3.796	165.13	35.93	1047.0	130.9	29.14	5.398	-	4690	-	5620	294.5
"	"	13.938	1.0313	164.86	66.05	"	3.649	152.6	48.49	1366.0	170.75	28.17	5.304	-	6445	-	7730	384.2
"	"	13.00	1.50	232.29	57.44	"	3.403	132.7	68.33	1816.0	227.0	26.58	5.154	-	9375	-	10250	510.8
18	18	17.50	.25	47.4	104.6	4.712	4.585	241.0	13.96	550.0	61.1	39.4	6.28	-	-	-	1670	137.5
"	"	17.375	.3125	59.032	102.5	"	4.549	237.1	17.36	679.3	75.47	39.12	6.254	-	1736	-	2080	169.8
"	"	17.25	.375	70.59	101.2	"	4.516	233.7	20.76	806.6	89.63	38.85	6.233	-	2080	-	2500	201.7
"	"	17.125	.4375	82.0	100.0	"	4.49	230.5	24.17	934.0	104.0	38.6	6.21	-	-	-	2920	234.0
"	"	17.0	.50	93.45	98.25	"	4.451	226.98	27.49	1063.0	117.0	38.31	6.19	-	2780	-	3330	263.3
"	"	16.874	.563	105.0	97.2	"	4.42	224.0	30.85	1177.0	130.9	38.1	6.17	-	-	-	3750	294.0
"	"	16.813	.5938	110.38	96.1	"	4.401	222.0	32.47	1233.0	137.0	37.97	6.157	-	3300	-	3950	303.3
"	"	16.75	.625	116.0	95.8	"	4.39	220.5	34.15	1290.0	143.2	37.8	6.14	-	-	-	4170	322.0
"	"	16.5	.75	138.18	92.55	"	4.32	213.82	40.64	1515.0	168.3	37.27	6.105	-	4170	-	5000	378.7
"	"	16.375	.8125	149.15	91.2	"	4.287	210.6	43.87	1625.0	180.56	37.04	6.083	-	4510	-	5420	406.2
"	"	15.686	1.1563	208.0	83.67	"	4.107	193.3	61.18	2182.0	242.33	35.67	5.969	-	6480	-	7760	545.5
"	"	14.625	1.6875	294.0	72.72	"	3.829	168.0	86.48	2909.0	323.22	33.64	5.798	-	9380	-	11250	727.2
20	20	19.25	.375	78.6	126.0	5.236	5.04	291.08	23.12	1113.0	111.3	48.16	6.940	-	1880	-	2250	250.5
"	"	18.875	.5625	116.77	121.1	"	4.941	279.8	34.35	1624.0	162.4	47.27	6.875	-	2810	-	3370	365.3
"	"	18.75	.625	129.33	119.5	"	4.909	276.12	38.04	1787.0	178.7	46.97	6.854	-	3120	-	3750	402.1
"	"	19.5	.25	52.7	130.0	"	5.11	299.0	15.52	759.0	75.9	48.8	6.98	-	-	-	1500	171.0
"	"	19.374	.313	65.8	128.1	"	5.08	295.0	19.36	937.0	93.7	48.4	6.95	-	-	-	1875	211.0
"	"	19.124	.438	91.5	125.1	"	5.01	288.0	26.95	1290.0	129.0	47.9	6.92	-	-	-	2625	290.0
"	"	18.25	.875	178.73	113.2	"	4.778	261.6	52.57	2409.0	240.9	45.82	6.769	-	4380	-	5250	542.0
"	"	18.188	.9063	184.81	112.7	"	4.761	259.8	54.36	2484.00	248.4	45.69	6.758	-	4530	4078	5440	558.9
"	"	17.438	1.2813	256.15	103.4	"	4.565	238.8	75.34	3317.0	331.7	44.03	6.634	-	6400	5770	7700	746.3
"	"	16.313	1.844	357.52	90.46	"	4.271	209.0	105.2	4489.0	448.9	42.67	6.543	-	9220	8300	11000	1010.0

[Concluded on next page]

Table of Properties of Pipe—concluded

Nom. Pipe Size	Outside Diam.	Inside Diam.	Wall Thickness	Wt. per Foot	Wt. of Water per Ft. Pipe	Sq.Ft. Outside Surface per Ft.	Sq.Ft. Inside Surface per Ft.	Transverse Area Sq.In.	Area Metal Sq.In.	Moment of Inertia	Section Modulus I/y	Radius Gyration Squared	Radius Gyration	ULTIMATE BURSTING PRESSURE* Butt-weld	Lap-weld	Hammer Weld	Seamless & Line Weld	Strength Factor Q 1000 Ft.Lb.
24	24	23.5	.25	63.4	189.0	6.283	6.15	435.0	18.67	1320.0	110.0	70.6	8.40	-	-	-	1250	247.5
"	"	23.376	.312	79.0	186.9	"	6.12	430.0	23.20	1630.0	136.0	70.1	8.38	-	-	-	1560	306.0
"	"	23.25	.375	94.62	183.8	"	6.087	424.6	27.83	1942.0	161.9	69.79	8.354	-	1560	1400	1880	364.2
"	"	23.125	.4375	110.1	181.8	"	6.054	420.0	32.39	2248.0	187.4	69.42	8.332	-	1820	1640	2190	421.6
"	"	23.00	.500	125.5	181.0	"	6.02	416.0	36.9	2550.0	213.0	69.0	8.31	-	-	-	2500	480.0
"	"	22.876	.562	141.0	178.5	"	5.99	411.0	41.4	2840.0	237.0	68.6	8.28	-	-	-	2810	534.0
"	"	22.75	.625	156.03	175.9	"	5.956	406.49	45.9	3137.0	261.4	68.35	8.267	-	2600	2340	3120	588.2
"	"	22.5	.75	186.24	172.1	"	5.892	397.61	54.78	3705.0	308.8	67.64	8.224	-	3130	2810	3750	694.8
"	"	22.125	.9375	240.0	166.4	"	5.792	384.45	67.93	4527.0	377.25	66.64	8.161	-	3900	3520	4690	848.8
"	"	21.938	1.0313	253.00	163.6	"	5.743	378.0	74.41	4920.0	410.0	66.12	8.130	-	4300	3870	5150	922.5
"	"	21.00	1.50	360.46	149.9	"	5.498	346.36	106.0	6727.5	561.75	63.59	7.973	-	6250	5630	7500	1264.0
"	"	19.625	2.1875	509.6	130.9	"	5.138	302.0	149.9	9010.0	750.83	60.11	7.75	-	9120	8200	10950	1689.0
26	26	25.5	.25	69.0	222.0	6.807	6.68	511.0	20.25	1680.0	129.0	83.0	9.10	-	-	-	1155	290.0
"	"	25.374	.313	86.0	220.0	"	6.64	506.0	25.27	2090.0	161.0	82.50	9.08	-	-	-	1450	362.0
"	"	25.25	.375	102.7	217.5	"	6.61	501.0	30.20	2480.0	191.0	82.0	9.06	-	-	-	1730	430.0
"	"	25.125	.4375	119.44	214.6	"	6.578	495.8	35.19	2871.0	220.8	81.7	9.039	-	1680	1520	2020	496.8
"	"	25.0	.50	136.17	212.5	"	6.545	490.9	40.06	3257.0	250.5	81.31	9.017	-	1920	1730	2310	563.7
"	"	24.874	.563	153.0	211.0	"	6.51	486.0	45.10	3660.0	281.0	81.0	9.0	-	-	-	2600	633.0
"	"	24.75	.625	169.4	209.0	"	6.48	481.0	49.8	4010.0	308.0	80.5	8.97	-	-	-	2880	693.0
"	"	24.625	.6875	185.86	206.2	"	6.447	476.3	54.67	4382.0	337.1	80.15	8.953	-	2640	2350	3170	758.4
"	"	24.5	.75	202.2	205.0	"	6.42	472.0	59.5	4760.0	366.0	80.0	8.94	-	-	-	3460	823.0
28	28	27.5	.25	74.0	256.5	7.33	7.2	595.0	21.8	2105.0	150.0	96.5	9.82	-	-	-	1070	338.0
"	"	27.376	.312	92.4	256.0	"	7.17	589.0	27.15	2600.0	186.0	95.9	9.79	-	-	-	1340	418.0
"	"	27.25	.375	110.6	254.5	"	7.14	586.0	32.6	3020.0	216.0	95.6	9.78	-	-	-	1610	486.0
"	"	27.125	.4375	128.79	250.1	"	7.101	577.9	37.88	3598.0	257.0	94.99	9.746	-	1560	1400	1875	578.3
"	"	27.0	.50	146.85	247.8	"	7.068	572.6	43.2	4085.0	291.8	94.56	9.724	-	1790	1610	2140	656.5
"	"	26.75	.625	182.7	244.0	"	7.01	562.0	53.8	5030.0	359.0	93.6	9.68	-	-	-	2680	808.0
"	"	26.5	.75	218.28	236.8	"	6.938	551.6	64.21	5964.0	426.0	92.89	9.638	-	2680	2420	3210	958.5
30	30	29.5	.25	79.3	297.5	7.854	7.73	685.0	23.4	2600.0	173.5	111.0	10.52	-	-	-	1000	390.0
"	"	29.376	.312	99.0	295.0	"	7.69	679.0	29.1	3210.0	214.0	110.2	10.5	-	-	-	1250	482.0
"	"	29.25	.375	118.0	292.5	"	7.66	673.0	34.9	3830.0	255.0	109.9	10.47	-	-	-	1500	574.0
"	"	29.124	.438	138.1	289.0	"	7.63	666.0	40.7	4560.0	304.0	109.2	10.45	-	-	-	1750	684.0
"	"	29.0	.50	157.53	285.9	"	7.592	660.5	46.34	5042.0	336.1	108.8	10430	-	1670	1500	2000	756.3
"	"	28.875	.5625	176.85	283.5	"	7.559	654.9	52.02	5637.0	375.8	108.4	10.41	-	1880	1690	2250	845.5
"	"	28.75	.625	196.1	282.3	"	7.53	650	57.7	6230.0	415.0	108.0	10.39	-	-	-	2500	934.0
"	"	28.5	.75	234.3	276.1	"	7.461	637.9	68.92	7375.0	491.7	107.0	10.34	-	2500	2250	3000	1106.0
32	32	31.0	.50	180.0	326.7	8.377	8.116	754.8	49.48	6162.0	385.13	124.5	11.14	-	-	1410	1875	866.5
"	"	30.75	.625	222.0	321.5	"	8.05	742.7	61.6	7601.0	475.06	123.4	11.1	-	-	1760	2340	1069.0
"	"	30.25	.875	305.0	311.1	"	7.919	718.7	85.56	10290.0	648.0	121.4	11.01	-	-	2460	3290	1461.0
34	34	32.75	.625	236.0	364.6	8.901	8.574	842.4	65.53	9132.0	537.58	139.4	11.8	-	-	1650	2210	1209.0
"	"	32.25	.875	323.0	353.6	"	8.443	816.9	91.06	12470.0	733.53	136.9	11.72	-	-	2320	3090	1650.0
36	36	34.75	.625	253.0	410.5	9.425	9.097	948.4	69.46	10900.0	605.55	156.9	12.51	-	-	1560	2085	1363.0
"	"	34.0	1.0	392.0	393.0	"	8.901	907.9	110.0	16890.0	938.33	153.5	12.38	-	-	2500	3340	2111.0

*Note: Butt-weld -- f = 40000 lb. per sq. in.
Lap-weld --- f = 50000 lb. per sq. in.
Hammer-weld--f = 45000 lb. per sq. in.
Seamless & Line-weld --
f = 60000 lb. per sq. in.

Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

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Performance of an Underfeed Domestic Stoker

By T. G. Estep* and D. C. Saylor** *(NON-MEMBERS)*

Pittsburgh, Pa.

THE particular stoker which is the subject of this discussion consists of a coal hopper from which fuel is delivered to a continuously driven screw. The screw forces the coal through the feed tube horizontally for approximately 3 ft. The pushing action of the feed screw then elevates the coal vertically into the retort. Surrounding the upper part of the retort is an air space with openings in the retort side through which air is supplied by a fan to the fuel bed for combustion.

A small amount of air is also supplied to the end of the feed tube to prevent furnace gases from blowing back through the coal hopper. Means are provided to prevent arching of the coal over the feed screw and the hopper is also fitted with a gas tight cover.

The air supply to the fuel bed is manually controlled by a louver damper in the fan inlet. A floating butterfly damper with a graduated index is installed in the fan discharge so that the proper supply of air can be approximately determined.

The stoker unit was mounted under a 1,200 sq ft capacity rectangular house heating boiler. The boiler was prepared for test work exactly in accordance with the recommendations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The water and coal supplied the unit were determined by weighing. The tests were started with the furnace hot and conditions constant. Carbon dioxide readings were recorded on a gas analyzer, and the flue gas composition checked with an Orsat apparatus. Steam pressure was maintained constant with an automatic back pressure valve.

The boiler used had a grate area of 8.2 sq ft when arranged for hand firing. When the stoker was installed, the area at the top of the retort was 0.492 sq ft. The top retort area was used in determining the pounds of coal per square foot of grate area since the air was distributed to the fuel bed mainly through this area. This neglected the slow burning of the coke thrown clear of the retort top. The furnace volume after the stoker was installed equalled 14.84 cu ft.

Two types of coal, a high and low volatile bituminous, were used in securing the results which are shown graphically in Figs. 1 and 2. The coal size was ⅜ x 1⅛, the size recommended by the stoker manufacturer. These coals had the analyses as given:

Proximate Analysis, As Received

	High Volatile	Low Volatile
Free moisture	1.8	2.0
Volatile matter	35.0	16.7
Fixed Carbon	52.8	74.0
Ash	10.4	7.3
	100.0	100.0

Ultimate Analysis, As Received

	High Volatile	Low Volatile
Hydrogen	5.10	4.51
Carbon	73.80	80.00
Nitrogen	1.40	1.76
Oxygen	7.40	4.73
Sulphur	1.90	1.50
Ash	10.40	7.50
	100.00	100.00
Heating value	13,260	14,500
Ash softening temperature	2,490 F	2,600 F

After several preliminary tests in which various furnace arrangements were used, it was decided to perform 3 series of tests with the furnace arrangement recommended by the stoker manufacturer. The tests were:

1. High volatile coal, no attention to the fire.
2. High volatile coal, attention to the fire.
3. Low volatile coal, no attention to the fire.

The results of the tests with attention to the fire are not included in the graphical representation of the results. The values of the various items that were obtained when the fuel bed received attention are given in the discussion, and will indicate the beneficial effect of fuel bed agitation in the performance of the stoker. The results obtained by breaking up the coke plug and raking the unburned carbon into the combustion zone are similar to the results which were obtained when working with a stoker that had positive fuel bed agitation.

When a domestic stoker is installed in a residence heating plant, its operation is controlled by various devices which maintain a reasonably constant temperature

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For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

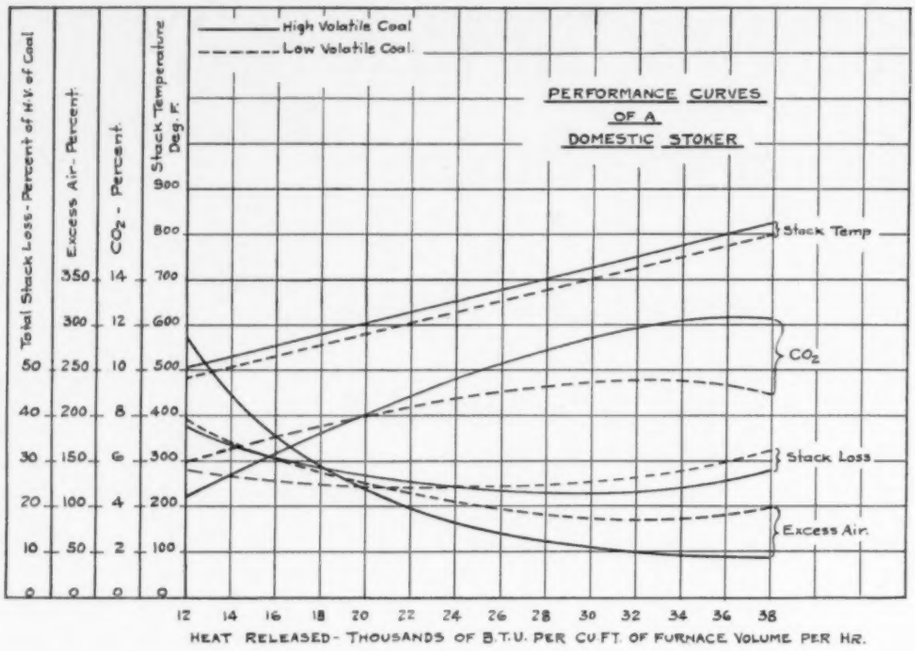
High Volatile Coal
Low Volatile Coal
PERFORMANCE CURVES OF A DOMESTIC STOKER
Total Stack Loss - Percent of H.V. of Coal
Excess Air - Percent
CO_2 - Percent
Stack Temperature Deg. F.
Stack Temp
CO_2
Stack Loss
Excess Air
HEAT RELEASED - THOUSANDS OF B.T.U. PER CU.FT. OF FURNACE VOLUME PER HR.

Fig. 1—Stack temperature, carbon dioxide, stack loss and excess air performance curves

in the dwelling by intermittent operation of the stoker. To determine the performance of the stoker under these conditions would require that the test be made in some residence and the duration would be for the entire heating season. Thus only one test could be made per year. By operating the stoker continuously at a constant rate it is possible to perform a large number of tests in a much shorter time and still determine the general operating characteristics, particularly the limitations. There is no doubt that this accelerated method of testing will magnify the faults of the stoker, but this may also be desirable.

The tests herein reported were made with the stoker operating continuously with a constant rate of feed during the test. The stoker is provided with 3 rates of feed and several tests were made at each rate so as to obtain an average performance.

Since there is no ash pit in a stoker installation of this type and since coke was formed which could not be burned, there is a question as to the ash pit loss. In the early tests an effort was made to remove the coke and ashes, to separate the ashes from the coke and to determine the amount of carbon in the ash. This was not a satisfactory procedure as the results of such determinations were not consistent. The method finally used was a heat balance method. The stack gas analysis, weight of water used, the coal analysis, and the stack temperature allowed the weight of coal actually burned to be computed. This computed weight of coal was from 1 to 8 lb of coal per hour less than the weight of coal actually supplied. Attention should be called to the fact that this represents a greater weight of carbon, since the volatile constitutents of the coal actually supplied the furnace were burned.

The performance of the stoker is represented by a series of curves shown in Figs. 1 and 2. The abscissa for each group of curves is the heat released by combustion, expressed in 1,000 Btu per cu ft of furnace volume per hour. The efficiency of the boiler is purposely eliminated from the results, as it was desired to obtain only the stoker performance and not the combined performance of the stoker and the boiler. Perhaps this somewhat limits the value of these tests to the heating and ventilating profession who generally judge the performance of the entire heating plant in terms of equivalent direct radiation and this unit of measurement does involve boiler efficiency. Since the efficiency is not constant over the range of operation shown, it is not possible to use a second scale of abscissa for the same curves showing equivalent direct radiation corresponding to a given heat release. However, if it is desired to estimate the equivalent direct radiation, if the given heat release is multiplied by an assumed boiler efficiency and the product divided by 240, the desired result will be secured.

Discussion of Results

The stoker manufacturer furnished a pair of tongs to remove the ash, the assumption being made that the ash would be fused in the form of clinker. The only clinker produced was a negligible amount at the top of the retort on the hopper side.

Both of the coals used had ash fusion temperatures that are within the range of the domestic fuel coals. Also the tests were run continuously which meant that hotter furnace conditions were obtained than on intermittent operation. If the ash does not clinker under the conditions of operation, then there is a dirty disagreeable task of ash removal which will upset the conditions of operation for a considerable time.

Where the coal is elevated vertically to the level of the fuel bed, the coal passage casting is formed with a long radius bend. The curve allowed the coal to pack on this side of the casting and this action extended up to the combustion zone. Air flowed through the fuel bed with less resistance on the side away from the curve. As a result of this difference in density of the fuel bed, the rate of combustion on the hopper side of the retort was much higher than on the opposite side, and the clinker which was formed occurred in this hot zone. The small

amount of clinker formed obstructed the air openings and tended to correct the uneven distribution of air. It was thought that by plugging some of the air openings on the hopper side the clinker formation might be avoided and that the air distribution would be more uniform from the beginning, but trials with some of these openings closed did not seem to improve combustion.

The screw for delivering the coal from the hopper to the retort operates continuously and at a constant speed for a given rate of feeding. Thus there is no shock or agitation to the fuel bed as there would be if the screw had an intermittent motion such as would be obtained if driven by a ratchet arrangement. As the coal passes through the retort and is partially burned, it produces a cylindrical coke plug, the diameter of which is slightly less than the diameter of the retort. This coke plug rises at an even rate of speed until it is approximately 8 in. above the top of the retort. The plug then breaks due to strains produced by cooling and some of the coke is thrown clear of the combustion zone. The coke thrown out of the fire zone does not burn readily, and, unless the fuel bed receives some hand manipulation, there are two distinctly different rates of combustion occurring in the furnace, one in the fuel remaining in the retort and the other in the fuel which has fallen aside.

The conditions mentioned previously influence the design of the fan materially for to attempt to force air through the plug will require a high head, while to supply air to burn the coke at the top of the air ring requires only a moderate pressure. The quantity of air supplied for a given rate of fuel feed should be constant but, since the fuel bed resistance is variable, a variable amount of air will be delivered to the burning coal.

If reference is made to the excess air curves, Fig. 1, it is seen that at low ratings there is a high percentage of excess air. The percentage of excess air decreases as the rate of burning coal increases and tends to become constant at the high values of heat release. This is due to the uneven packing action of the screw feed and is especially noticeable when there is a light fuel bed. As the fuel bed becomes heavier with the increased rate of coal feeding, there is a better air distribution. If the fan forces more air through the packed heavy fuel bed than when the fuel bed is less dense, too much air will be supplied. This action will sometimes cool the coke plug and an accumulative effect results.

When the fuel bed was manipulated by hand, the excess air was materially reduced as was shown by the higher CO_2 content of the products of combustion of about 12 per cent, which indicates the effects of a more uniform fuel bed.

The CO_2 content of the flue gases is often cited as a criterion of the completeness of the combustion process. Since there was no CO present, it is apparent that the coal consumed was burned completely, but it does not indicate anything about the coke thrown clear of the combustion zone. The CO_2 curves for the two coals show that at low rates of combustion the low volatile coal gave a much higher CO_2 content than the high volatile. The low volatile coal was a non-caking variety and gave a fairly uniform fuel bed at all times. The high volatile coal was a caking type and at low rates of combustion most of the air was blown through the fissures formed in the fuel bed and was not used in the combustion process. At the higher rates of combustion the CO_2 content from the high volatile coal was more than the low volatile for two reasons; first, the volatile matter rising above the fuel bed utilized some of the excess air in the furnace atmosphere; and, second, the low volatile coal, being a slow burning coal, required a large amount of air pressure to maintain the proper combustion rate. It would seem that for burning high volatile caking coals the fan should produce a relatively high head with a corresponding small volume, but for low volatile, non-caking coals the volume should be large with a small head.

The stack temperature curves are essentially straight lines and the values do not exceed those of good practice.

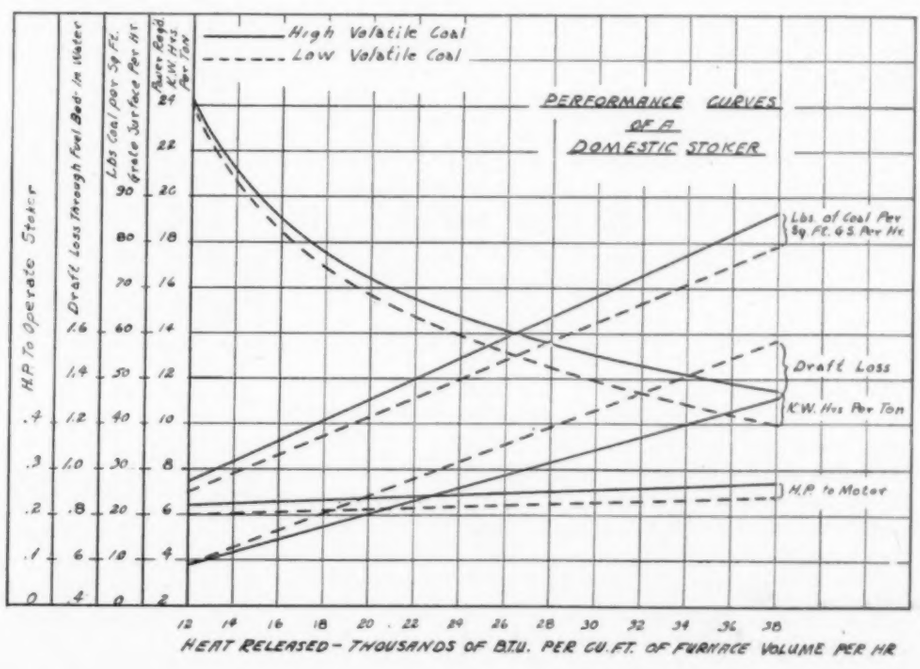
High Volatile Coal
Low Volatile Coal
PERFORMANCE CURVES OF A DOMESTIC STOKER
Lbs. of Coal Per Sq. Ft. G.S. Per Hr.
Draft Loss
K.W. Hrs Per Ton
H.P. to Motor
H.P. To Operate Stoker
Draft Loss Through Fuel Bed-In Water
Lbs Coal per Sq. Ft. Grate Surface Per Hr
Power Reqd. K.W. Hrs Per Ton
HEAT RELEASED - THOUSANDS OF B.T.U. PER CU. FT. OF FURNACE VOLUME PER HR

Fig. 2—Coal rate, draft loss and power consumption performance curves

The curves are the results of two actions: namely, the heat released in the boiler and the heat absorbed by the boiler heating surfaces from the hot gases. The higher stack temperatures obtained with the high volatile coal are due to the longer flame.

The high volatile coal gave generally a lower stack loss than the low volatile coal. The loss is a product of the temperature difference and the weight of the gaseous products of combustion. While the stack temperature was higher with the high volatile coal, the weight of the products of combustion was enough lower to reduce the total loss below that of the low volatile coal.

The stack losses were lowered about 5 per cent when the fire was hand manipulated. This was due to the fact that manipulating the fuel bed resulted in a glowing mass of fuel. This glowing mass radiated more heat to the furnace than the top of the coke plug, and also reduced the excess air. The stack loss curves include the loss due to moisture in the coal and also the moisture formed from the combustion of the hydrogen in the fuel bed.

The power supplied the unit was almost constant, increasing slightly at the increased rates of coal feed. Most of the power supplied the unit was used in overcoming friction. For this reason the power per ton of coal was greater at the low rates of coal feed. The draft loss through the fuel bed was greater for the low volatile than for the high volatile coal. This may be accounted for by the fact that the low volatile coal, being very friable, broke up into a greater proportion of fines which gave a more compact fuel bed. The high volatile coal, because of its caking properties, produced large fissures in the fuel bed which offered little resistance to the flow of air.

Conclusions

1. The stoker is satisfactory in its operation provided a non-caking variety of coal is used. The coal should have an ash fusion temperature low enough to fuse the ash so it can be readily removed.
2. Caking coals will be hard to manipulate with this stoker.
3. From the results of these tests and other tests, it is evident that the action of the stoker may be improved by the proper design of fan.
4. The long radius bend in the coal feed passage is responsible for faulty air distribution in the fuel bed.
5. If air could be introduced into the center of the fuel bed, the probability of the coke plug forming would be lessened.
6. The best results were obtained when hand attention was given to the fuel bed.

Solar Radiation Measurement

Previously the valuation, in absolute units, of the energy in the short wave length ultraviolet region of the solar spectrum has been attempted by several methods. The most common procedure has been to integrate the energy curve observed throughout the whole solar spectrum and calculate that fractional part which is contained in the band of wave lengths under investigation. Then from a determination of the total radiant flux that portion in the short wave lengths is calculated. The principal objection to this method in the past has been the lack of accurate data on the spectral-energy distribution and particularly in the ultraviolet region.

A proposed variation of this method of calculation is referred to in a recent article entitled, Evaluation of Ultraviolet Solar Radiation of Short Wave Lengths, by W. W. Coblentz and R. Stair which appears in the April, 1936 *Journal of Research* of the National Bureau of Standards. In this paper technical details are given for evaluating ultraviolet solar radiation by two closely agreeing methods: (a) by means of a balanced thermopile and filter radiometer, calibrated against a standard of thermal radiation, and (b) by means of a photoelectric ultraviolet meter calibrated against a standard of ultraviolet radiation. A photoelectric cell and filter method for determining the solar ultraviolet spectral-energy distribution is described. Data are given on ultraviolet solar radiation intensities observed at Washington, D. C., San Juan, P. R., and Flagstaff, Ariz. The intensity of ultraviolet solar radiation of wave lengths shorter than and including 3132 A, in the clearest, midsummer, midday sunlight at Washington is about 75 micro watts per square centimeter decreasing to one-tenth this value (8 micro watts per square centimeter) during the clearest, midwinter, midday sunshine. Extrapolation of the data observed at the three stations (Washington, San Juan, and Flagstaff) indicates an ultraviolet intensity of about 600 micro watts per square centimeter, outside the earth's atmosphere—a fivefold to eightfold increase as compared with a 20 to 30 per cent increase in total intensity of all wave lengths.

Symposium on Industrial Fuels

A Symposium on Industrial Fuels recently published by the *American Society for Testing Materials* comprises four extensive technical papers as follows: Coal and Coke: Occurrence, Testing and Utilization, by A. C. Fieldner, Chief Engineer, Experiment Stations Division, U. S. Bureau of Mines, and W. A. Selvig, Supervising Chemist, Miscellaneous Analysis Section, Pittsburgh Experiment Station, U. S. Bureau of Mines; Industrial Fuel Oils, by H. V. Hume, Combustion Engineer, The Atlantic Refining Co.; Manufactured Gas, by P. T. Dashiell, Vice-President, and F. H. Trembly, Jr., Assistant Sales Manager, The Philadelphia Gas Works Co.; Liquified Petroleum Gas, by W. H. Bateman, President, Salgas, Inc.

At the suggestion of the *A.S.T.M.* Philadelphia District Committee, which with representatives from the Engineers' Club planned the symposium, the authors followed a prescribed outline so that the papers have considerable comparative value. Among the subjects covered are the following: historical background, magnitude of the industry, future availability of raw materials, possibility of new processes, tests which are applied to the materials, significance of tests, utilization of materials, general economic aspects of the product and utilization.

Copies of this publication, containing 70 pages, in heavy paper cover, can be obtained from *A.S.T.M.* Headquarters, 260 S. Broad St., Philadelphia, Pa., at 75 cents per copy, or on orders for ten or more copies, at 60 cents each.

The Distribution of Steam in Heat Transfer Surface

By **John McElgin*** ***(MEMBER)***, **Philadelphia, Pa.**

DURING the past 10 years copper fin and tube heating surface has demonstrated its effectiveness for the transfer of heat from steam to air and is today generally accepted as a universal standard in forced air heating systems. While a wide variety of designs are in existence, the principal differences arise in structural details and the ratios between prime and extended surface. In general, it may be said that available commercial designs have the following characteristics in common:

(1) Light weight and compactness.
(2) A high condensing rate per unit of weight and face area.
(3) Low resistance to air flow.
(4) Flexibility in installation permitting the field assembly of individual sections to obtain the desired temperature rise and the air flow area.

These features, are, of course, widely recognized and require no further discussion here.

For the most part investigations and developments in the field have been purely on the basis of maximum heat transfer—the principal aim being to arrive at a combination of fin and tube areas, resistance to air flow and a condensing rate that leads to the most effective use of the material employed.

However, the widespread use of automatic temperature control and the reliance that the modern air treating system must necessarily place on such equipment have created a new problem in the design and application of fin and tube surface. It is an outstanding fact that in the usual system the heating surface is only rarely called upon to deliver its full capacity, but during the greater part of the time operates well below the maximum. To fulfill these requirements it is desirable that the heating surface possess not only a high condensing capacity but that it also readily lend itself to systems of continuous control at the lesser demands.

It is the purpose of this paper to describe a means of securing a close coordination between fin and tube steam heating surface and automatic temperature control in all stages of operation. Briefly, this is accomplished by inserting within the usual condensing tubes additional tubes which serve the purpose of distributing steam uniformly along the length of the condensing tube. While this simple change would not appear to greatly alter surface characteristics it can readily be demonstrated that steam distributing tubes impart a high degree of flexibility to the usual design. The several features that are described here are the combined result of improved steam distribution and a control system capable of intermediate action.

Automatic control of the heating effect of fin and tube surface or for that matter any type of steam heating element may be obtained in two distinct ways namely:—(1) by-passing of air and (2) modulation of the steam supply.

A comparison of the two methods on the basis of

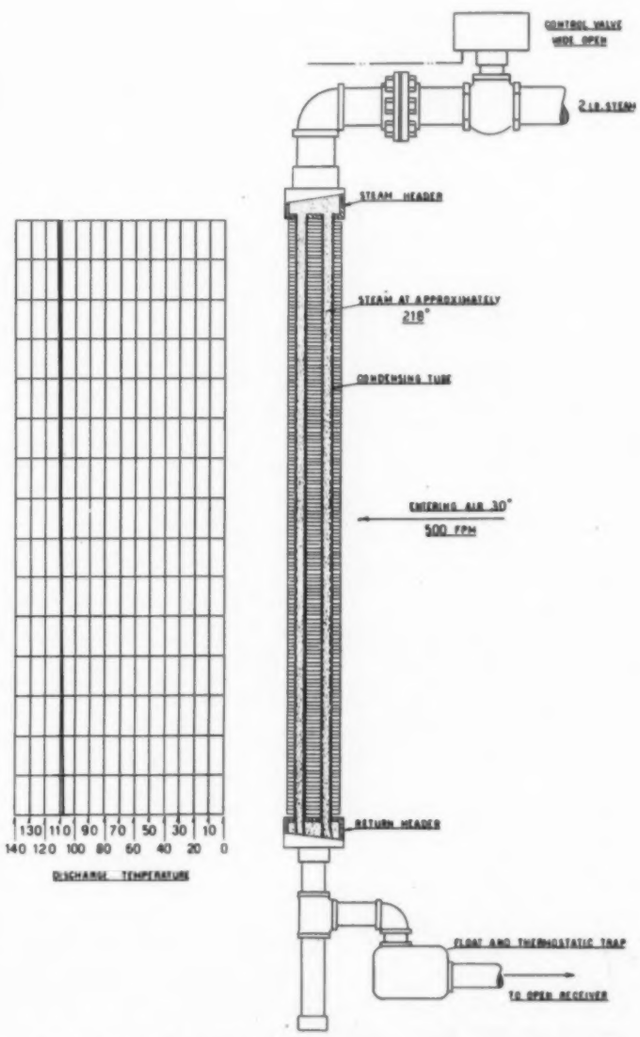

Fig. 1—Chart of discharge temperatures and section through heating surface with steam supply valve wide open

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For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

simplicity and economy indicates advantages for the valve control arrangement. All by-pass dampers and channels are eliminated and the air flows constantly through the heating surface—resulting in a saving in space and first cost and an air volume that is generally unaffected by the thermostatic system. Full control of the surface from maximum capacity to positive shut off is accomplished by a single throttling valve that is subject to proper design and sizing prior to installation.

One of the disadvantages in the by-pass method is the separation of the air handled into hot and cold air streams. Unless a strong mixing action is available on the discharge side of the surface this temperature stratification will frequently persist even in long runs of duct. With streams of air at widely separated temperatures naturally no single point can be selected as representative and close air stream control is a practical impossibility. A typical example of this condition arises where the by-pass principle is used on preheater sections for comfort or process humidifying systems.

If provisions are not made within the condensing tubes to provide for steam distribution, a similar degree of temperature stratification will occur when the supply of steam to fin and tube surface is throttled. When the volume of steam supplied is less than the quantity that can be condensed by the surface, by-passing will be created at the return end. This effect is aggravated by extending the air side area to the practical limit and arises from the high rate at which heat is transferred from steam to metal walls.

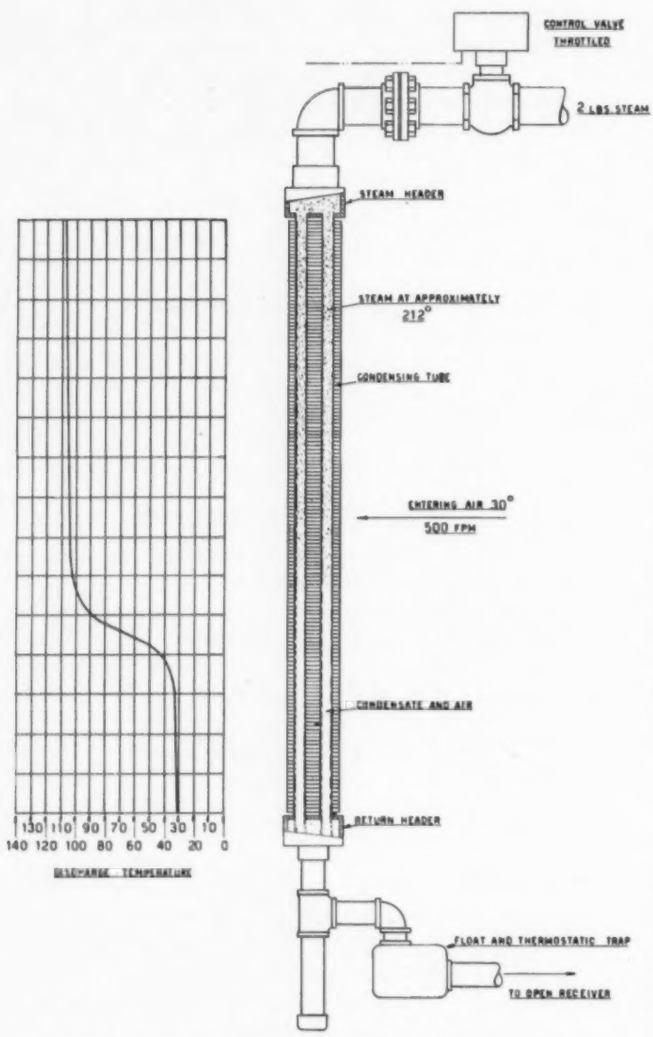

Fig. 2—Chart of discharge temperatures with throttling steam supply to heating surface

The condition may be illustrated by considering a typical example:—Fig. 1 shows a section of surface connected to a steam pressure source of 2 lb per sq in. with the condensate draining to an open receiver through a float and thermostatic trap. This corresponds to the usual arrangement on a pump and receiver atmospheric return system. For the purpose of illustration, the section will be assumed to have the capacity to raise the air at 500 fpm face velocity from 30 to 110 F at a 2 lb steam pressure.

Initially with the valve wide open the condensing tubes will be filled with steam at approximately 218 F. This is assuming an oversized valve and a negligible resistance to flow in the condensing tubes. As the supply valve is throttled a point will be reached where the condensing tubes are just filled with steam at atmospheric pressure at approximately 212 F. A further restriction of the valve will decrease the volume but not the temperature of the steam supplied. The lowest available steam temperature will, in all cases, correspond to the pressure conditions maintained in the return line.

The reduction in capacity effected by the change in steam pressure and temperature conditions within the condensing tubes may be calculated from the relation of steam to entering air temperatures.

$$\text{Reduction in capacity} = 100 - \frac{212-30}{218-30} = 3.5 \text{ per cent.}$$

The major reduction in capacity is obtained after the pressure and temperature of the steam have been reduced to the lowest possible level and it is in this range that surface under continuous automatic control will operate the greater part of the time. As will be apparent from the discussion herewith it is also the region in which the problem of steam distribution becomes a most important consideration.

Continued throttling of the supply valve shown in Fig. 1 eventually produces the condition shown in Fig. 2.

At the steam end the condensing tubes are in full contact with 212 deg steam and that portion will operate at 96.5 per cent of its former capacity. A similar section at the return end will have practically no heating effect since the condensing tubes contact only condensate and air. With the surface under automatic control the heated portion increases or decreases in accordance with the demand—hence, the comparison with by-passing of air.

In view of the danger of freezing the condensate at the return end, continuous valve control is not generally attempted in the presence of sub-freezing air. It is common practice to protect controlled preheaters by additional sections designed and operated to maintain the entering air above 32 deg. Usually two such sections are used and are controlled by outdoor thermostats adjusted to fully open the respective steam valves at 35 F and at 0 F.

While in the past both of the previous methods have, under many favorable circumstances, proved adequate, it should be recognized that in at least an equal number of

cases a wide variation of air stream temperature has been a serious hindrance to the best operation of control equipment. Further the step action necessary on valve controlled preheaters is open to the objection of increased cost and operation that bears only an approximate relation to process demands.

One method of solving the problem of a variable discharge temperature may be obtained through the use of steam distributing tubes. As the name implies, these tubes serve to conduct the steam axially along the condensing tube releasing it at fixed intervals and in practically equal amounts regardless of the quantity supplied. The net effect is to provide a gradual increase or decrease of the whole air stream temperature in accordance with the demand. Coupled with a well designed system of continuous control this arrangement exhibits another equally important characteristic: it permits throttling of the steam supply in the presence of sub-freezing air without danger of tube breakage due to freezing of the condensate.

Naturally, there are limits to the degree of throttling for there exists no method of protection that will prevent freezing when the volume of steam is permitted to fall to abnormally low levels. In order to derive benefit from this method of steam introduction it is essential that an air stream thermostat be located in the path of the air leaving the preheater section and adjusted to keep the discharge temperature and hence the volume of steam supplied above a safe minimum.

The typical effect of throttling the supply of steam to a section of fin and tube surface equipped with steam distributing tubes is shown in Fig. 3. This is a condition comparable to that previously shown in Fig. 2 and the same steam pressure and temperature conditions will prevail. However, in this case steam entering the supply header is forced to pass first through the distributing tube and thence through the orifices to contact the condensing tubes.

With a limited volume of steam the quantity issuing from each of the orifices is condensed in a small outer tube area opposite the jet, producing a series of active sections along the total length of the surface. This does not, however, create a similar series of hot and cold streams for both the high conductivity of the condensing tube and a close spacing of the outlets equalize the temperature distribution. The distribution shown in Fig. 3 is typical of temperature measurements made in the air stream 6 in. from the face of the surface.

When the steam quantity is increased, the active condensing tube area increases to a point where the tube is filled with steam and the surface is operating at maximum capacity for the entering air and steam conditions prevailing.

A comparison between Figs. 2 and 3 shows that steam distributing tubes are a step towards solving the problem of variable discharge temperatures. It is not the intention, of course, to create the impression that this device is capable of producing absolute uniformity, for this appears from experience to be beyond the limits of practice. It is possible, however, by careful regulation of the orifice sizes and inner tube areas to maintain a comparatively negligible difference between any two points in the air stream.

As an example, the type of two-row section illustrated in Fig. 3 will have the capacity to heat air from 30 to 110 F at 500 fpm and a 2 lb steam pressure. With the supply valve throttled, the average variation in discharge temperature over the face of the section with uniform air flow will vary over a range from 5 to 10 F, depending upon the length of the condensing tube. In direct contrast the same section without distributing tubes, will in some cases under less than maximum demand conditions separate the air volume into air streams that may vary approximately 70 F apart over the entire surface.

The method used in the design of distributing tubes is largely one of trial and error based on tests of the particular section at hand. As would be expected the major controlling factors are the finned length and the diameter of the condensing tubes. It is obvious that long sections require more careful attention than short ones of equivalent fin spacing and capacity per unit of face area. The diameter or the internal area of the condensing tube directly affects the space available for steam distribution and therefore limits the maximum length of finned tubing. In general, surface designs employing condensing tubes 5/8 in. or over are more readily adapted to the

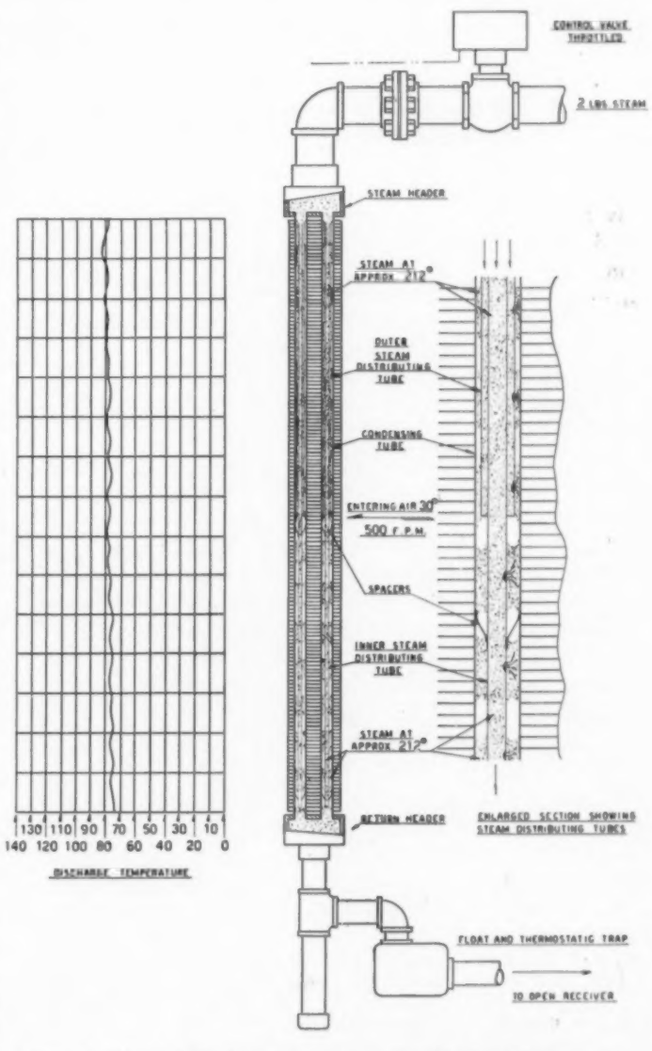

Fig. 3—Chart of discharge temperatures with throttling steam supply to heating surface using concentric steam distributing tubes

principle of steam distribution than ones with smaller tubing.

The original design of distributing tube was based on the use of a single tube extending the full length of the condensing tube. This construction was found to have several disadvantages chief among which was the practical limitation of finned tubing for good distribution to approximately 3 ft. An arrangement adapted to lengths up to 6 ft with ¾ in. condensing tubes and capable of much better steam distribution was that of employing multiple tubes all of which receive steam directly from the primary header. The design shown in Fig. 3 employs two concentric tubes of 9/16 in. and ⅜ in. diameter inside a ¾ in. condensing tube. The first or larger tube serves to supply one half of the surface and the second tube the remainder.

A feature of this construction is that the division of flow occurs at the header and the relative quantities supply both ends if subjected to close control.

From a control standpoint the practical importance of the more uniform air stream temperature obtained with the distributing tubes is dependent upon the location and the function of the heating surface. Some applications are naturally more sensitive in this respect than others, the most critical being those where close control is desired from a thermostat located directly in the path of the air leaving the surface, and there is no opportunity for even partial mixture of the streams. An example of this condition is in preheater sections for humidifying systems where the proper location of the dewpoint thermostat has frequently been a problem. Booster heaters fall under much the same heading. An arrangement less sensitive to temperature stratification is that of mixing the air in a single fan. It should be noted, however, that passage through a fan does not always produce a complete mixture since this will depend upon the temperature difference and the location of the surface with respect to the fan inlet or inlets. Many unit designs employ multiple fans with the surface under steam control and the condensing tubes parallel to the shaft axis. In this case, practically no benefit is derived from mixing and the fans handle air at widely different temperatures. Here and in similar instances steam distributing tubes will usually be found to be of value in reducing the temperature stratification to a minimum.

Perhaps the outstanding feature of this device as applied to fin and tube heating surface is the inherent protection offered against freezing when the supply of steam is throttled. This arises directly from the fact that the reduced volume of steam is brought into contact with a series of areas distributed along the length of the condensing tube rather than the whole area concentrated at one end. The net result is that a preheater under continuous control does not require additional protective sections but may be safely throttled within wide limits in the presence of subfreezing air.

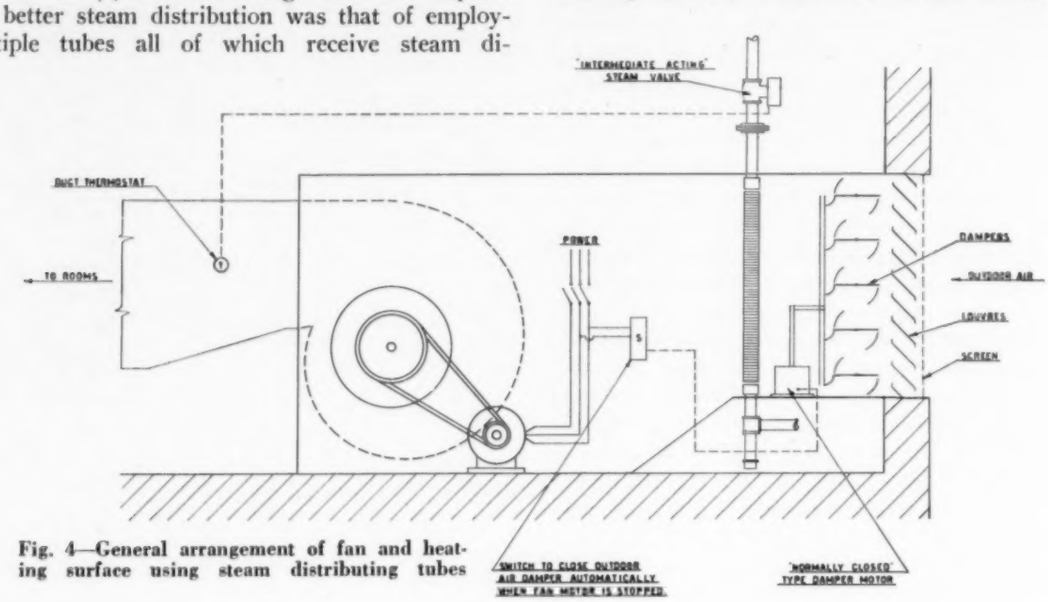

Fig. 4—General arrangement of fan and heating surface using steam distributing tubes

As indicated previously this arrangement cannot be regarded as a universal cure for any and all freezing problems but will have advantages during operating periods where the steam supply is continuous and where a low limit thermostat adjusted for 45 F or above is employed on the discharge side of the surface. Actual protection is then accomplished in the automatic control equipment and the distributing tube simply insures that all parts of the condensing tube will receive a representative quantity of steam. It should be observed that with the discharge temperature fixed safely above the freezing point the demand for steam increases as the outside temperature falls and the surface approaches towards maximum capacity. This automatically provides against extremely close throttling in cold weather. The use of a low limit thermostat does not impose a special control requirement, for this type of instrument usually forms an important part of every automatically controlled ventilation system.

Fig. 4 illustrates a simplified arrangement possible when a heating surface equipped with distributing tubes is used to preheat outdoor air. While this is shown for the usual tempered air ventilating system it serves as an example for other preheater applications since the only change required is the relocation of the duct thermostat. It should be noted in this arrangement that the usual positive acting preheaters are omitted and full control is accomplished with the thermostat located in the leaving

air stream. Normally a two-row section of the average design will meet this requirement but where an additional section is necessary this should be separately valved and operated in sequence with the first section.

Throughout the previous discussion use has been made of the term *continuous steam control*. By this is meant a combination of a steam valve and a thermostat so selected that the supply of steam is exactly the right quantity to meet the thermostatic demands without hunting. In view of the fact that this type of operation is essential to the best performance of any air heating system it will be well to observe the conditions under which it is possible.

The control that has been found most suitable for steam valve operation is of the intermediate acting or proportioning type in which the valve assumes a fixed position for various temperatures affecting the thermostat. The temperature change required to cause full travel of the valve is termed the *thermostat differential*.

The use of proportioning control in principle is, however, never sufficient to insure continuous results in the field. This will require that careful attention be given: (*a*) thermostatic differentials; (*b*) valve design and size; and (*c*) operating steam pressures. All three factors are very closely related and should be considered in combination to obtain the desired results.

In general, widening of the thermostatic differential will tend toward more gradual control and at the same time a system less sensitive to changes in demand. In view of the latter it is desirable to employ as narrow a differential as possible although a slight change in control point with the demand must always be accepted if hunting is to be avoided. Normally, room type thermostats have a differential of 3 to 6 F whereas the differential on duct type thermostats is three or four times this value. This does not imply a lowering of the control point by this amount, for a wide open valve may not be necessary to meet the maximum demand. This will depend on the valve size and the working steam pressure since a reduction in either will produce much the same result as widening of thermostat differential. The usual shift in the control point for outdoor air preheater applications falls in the range from 3 to 6 F.

Attempts to reduce this by increased valve size or decreased ductstat differentials generally result in hunting. A practical means of securing as close a differential as possible is by employing instruments whose differentials are capable of readjustment at the job.

To function properly under close automatic control a valve must offer considerable resistance to the flow of steam even in the wide open position. This is verified by the fact that the major reduction in surface capacity is obtained after the pressure in the condensing tubes has been reduced to a point near the return line pressure which may be either atmospheric or vacuum. With the surface under close control the pressure drop through the valve is practically equivalent to the differential from steam main to return main excepting, of course, the slight resistance to flow in the condensing tubes. If the valve is oversized, and this appears to be the usual tendency, control over the surface capacity is concentrated in a small percentage of the total movement causing the system to be unstable. To some extent an oversized valve can be compensated for by increasing the thermostat differential, but the better method is to employ a smaller valve particularly if two valves are to be operated in sequence from one instrument.

An effective method of sizing control valves is on the basis of resistance to flow of steam under maximum conditions and with recognition of the line pressure employed. For example, on a 2 lb steam pressure system wide open, a valve having a resistance under maximum demand equal to from 50 to 75 per cent of the initial pressure will give good results. As the initial steam pressure increases to 10 lb the pressure drop through the valve should fall in the range 70 to 80 per cent of the initial pressure. Higher line pressures will require correspondingly greater pressure drops. The principal consideration is to have the valve as close to the major control range as possible so that any valve movement will produce a corresponding change in air stream temperature.

Some attention should also be given to the design of control valves for modulated results with the view of obtaining an approach to straight line lift-flow characteristics. This may be approximated through the use of globe valves equipped with V ports or other similar throttling devices. The tendency of a standard flat disc valve to have its control range concentrated in a small initial lift renders it generally unsuited for this work.

As may be inferred from the above explanation, the matter of working pressures is intimately connected with that of valve sizing. Experience has proven that the most practical range of steam pressures for close control is 2 to 10 lb. This is because the heating surface under steam control operates in a low pressure region and the valve must function with the necessary differential. Where high pressure steam is employed the problem of sizing becomes quite critical and careful attention must be given the erosion resisting qualities of the disc and seat materials.

Summary

1. Several of the difficulties encountered in the application of automatic temperature control to fin and tube steam heating surface are a direct result of the temperature stratification created when air is by-passed or when the steam supply is modulated. In the first the condition is generally fundamental to the process, while in the second it is due to complete condensation in a limited part of the section nearest the supply.

2. In a system employing fin and tube surface under valve control it was found that temperature stratification may be reduced by inserting steam distribution tubes within the usual condensing tubes. These tubes serve to distribute reduced volumes of steam over a maximum length of condensing tube and so effect a practically uniform discharge temperature.

3. This arrangement inherently resists freezing of the condensate in the heating surface. The steam supply may be throttled within wide limits in the presence of sub-freezing air, making it possible to eliminate the positive acting preheaters usually installed to protect the controlled preheater. The conditions under which this operation is possible are simply that the steam supply be continuous and that the volume be maintained above a low limit by an air stream thermostat installed in the path of the discharged air.

4. Any system of proportioning steam valve control as applied to fin and tube surface may be considerably enhanced by taking the proper precautions to prevent *hunting*. Aside from the type of control the factors that determine whether or not the operation is free from this defect are: (*a*) thermostat differential; (*b*) valve size; and (*c*) operating steam pressures. Recognition of their influence on the control system will materially improve the final results.

A Field Study of the Heat Requirements of a College Building

By **F. E. Giesecke* and W. H. Badgett** *(MEMBERS)*, College Station, Texas**

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Texas Agricultural and Mechanical College

IN cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the Texas Engineering Experiment Station of the Agricultural and Mechanical College of Texas conducted a field study of the heat requirements of the Petroleum Building at the college, during the heating season of 1935-1936.

The building used for these tests, shown in Fig. 1, houses the Texas Engineering Experiment Station, the Department of Petroleum Engineering, and the Department of Geology. The building has a reinforced concrete frame and walls of brick, tile, and cast stone. Its cubage is approximately 522,000 cu ft and, with respect to heat storage capacity, its mass is approximately equivalent to 1,700,000 lb of water, not including the tower or the foundations. The tower houses the expansion tank for the central hot water heating plant which supplies heat to 38 major college buildings. The expansion tank has a capacity of 8,000 gal.

The building under test is heated by a reversed-return hot water system having 9,500 sq ft of equivalent direct cast iron radiation. Heat is supplied to the building continuously, day and night, except during those periods when the outdoor temperature is above 65 deg. Water for the system is supplied from the central mains in the heating tunnel located in front of the building. By employing the valve and pump equipment shown in Fig. 2, the hot water from the power plant is mixed with cooler water from the return main in the building to secure the desired flow temperature in the system, and then pumped through the various radiator circuits in the building. This mixing and recirculating arangement is employed to permit the use of higher temperature water in the tunnel flow mains and a correspondingly smaller volume of water, and to provide for the variations in the individual requirements of the many buildings heated from the central plant. Once a satisfactory setting of the mixing valves is obtained for a building, it is not disturbed for the remainder of the heating season. The water temperature in the flow mains of the tunnel is controlled at the power plant and is varied according to the outdoor temperature as shown in the accompanying schedule:

OUTDOOR TEMPERATURE	FLOW MAIN WATER TEMPERATURE	RETURN MAIN WATER TEMPERATURE
25	240	180
30	221	168
35	202	156
40	183	143
45	165	131
50	146	119
55	127	107
60	109	94
65	Heat Discontinued	...

Electrical recording and integrating flow meters were installed in the supply and return mains of the building, as shown in Fig. 2, to determine the quantity of water which passed through the heating system of the building. These meters were calibrated in the laboratory of the Texas Engineering Experiment Station before they were installed and their percentages of error over the entire operating range were found to be less than plus or minus 2 per cent. After installation, the operation and accuracy were checked by a factory representative. The temperatures of the water in the supply and return mains

Fig. 1—Texas Engineering Experiment Station, Petroleum Engineering and Geology Building, Agricultural and Mechanical College of Texas

*Director, Texas Engineering Experiment Station.
**Research Assistant, Texas Engineering Experiment Station.
For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

were registered by a two-pen recording thermometer which was very frequently checked against calibrated indicating thermometers located in the mains adjacent to the recording bulbs (Fig. 2). From the data provided by the recording flow meters and the recording thermometers the quantity of heat dissipated by the heating system of the building during any period of time could be readily calculated.

Outdoor temperatures were secured from a recording thermometer located in a standard United States Weather Bureau thermometer shelter 100 yards north of the building. This instrument was checked at frequent intervals with a Bureau of Standards calibrated thermometer.

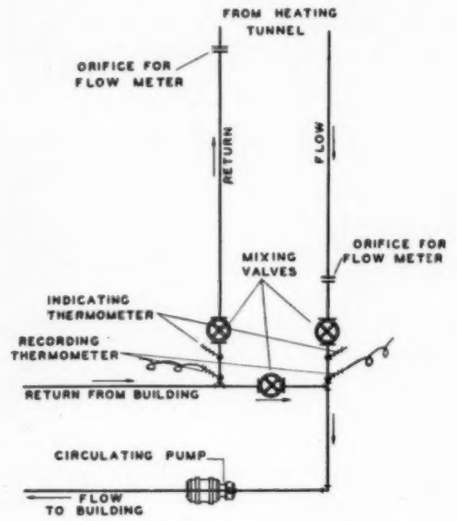

Fig. 2—Method of mixing and recirculating water

Indoor temperatures were taken at representative points throughout the building from 30 wall-type indicating thermometers and 10 recording thermometers placed 5 ft above the floor, on the inside walls and away from windows.

Wind velocities were recorded by an electrical wind velocity recorder of the U. S. Weather Bureau pattern with a standard three-cup anemometer located 9 ft above the roof of the building at an unobstructed point. The direction of the prevailing wind was determined by observing an indicating weather vane also located on the roof. With the exception of the south side, the building was not sheltered in any manner from the effect of the wind.

In computing the heat requirements of the building, the procedure and recommendations in the 1935 edition of THE A. S. H. V. E. GUIDE were followed with extreme care. From studies of temperature and wind velocity records of the U. S. Weather Bureau observation station at the college, an outdoor design temperature of 25 F and a wind velocity of 15 mph from north to northwest were selected as meeting THE GUIDE requirements.

An indoor design temperature of 70 F was used for the entire building except for laboratories of a shop nature and for halls for which a temperature of 65 F was used. The temperature beneath concrete floors on a fill was assumed to be 5 deg below the room temperature.

In the earlier calculation, the air temperature beneath the floor slab suspended above the ground was assumed to be 50 F. After several weeks of observations, a recording thermometer was placed in several locations beneath the slab for a period of two weeks. During this time the outdoor temperature range was from 15 F to 65 F, whereas, the air temperature beneath the floor slab varied only from 69 F to 71 F. The heat required to maintain a temperature of 70 F beneath the floor slab was supplied by the insulated flow and return mains, and by the riser branches beneath the slab, and should, therefore, be included in the estimate. Consequently, the heat losses for the foundation walls enclosing the space beneath the suspended floor slab were calculated and substituted for the previously determined losses through the floor slab.

No empirical exposure factors were used to increase the calculated heat losses on the sides of the building exposed to the prevailing winds. Air leakage was determined by the crack-infiltration method from the values given in Table 2, Chapter 6, of THE 1935 GUIDE.

For the conditions outlined above, the heat requirements of the building were estimated to be 1,399,000 Btu per hour, or 31,750 Btu per hour, per degree difference, indoor to outdoor temperature.

The results of calculations for observations made during seven complete 24-hour periods, from midnight to midnight, in January and February, 1936, indicate that the estimated heat requirement averaged about 23 per cent higher than the actual heat requirements of the building. The estimated requirement was higher than the actual requirement for the seven 24-hour periods by the following percentages: 24.6, 28.7, 29.2, 21.4, 14.9, 28.8, and 17.2 per cent.

The general character of these 24-hour observations is shown in Table 1, which contains the observations for the 24-hour period of January 27, 1936.

Table 1—Observations from Midnight to Midnight, January 27, 1936

Average Building Temperature 73 F

Hour of Day	Outdoor Temp. F	Degree Hours	Water Temperatures F			Lb Water Flowing per Hour	Heat Supplied Btu	Wind Vel. Mph and Direction	General Weather Condition
			Flow	Return	Difference				
A. M.									
M–1	37.1	35.9	191.0	143.0	48.0	19,500	936,000	15N	Cloudy
1–2	37.0	36.0	192.5	144.0	48.5	19,920	966,120	15N	Cloudy
2–3	36.8	36.2	194.0	146.0	48.0	20,520	984,960	15N	Cloudy
3–4	35.8	37.2	196.7	148.0	48.7	21,120	1,028,544	15N	Cloudy
4–5	34.0	39.0	202.8	150.0	52.8	21,000	1,108,800	11N	Cloudy
5–6	32.7	40.3	206.4	153.0	53.4	20,820	1,111,788	9N	Cloudy
6–7	32.6	40.4	211.0	155.0	56.0	20,880	1,169,280	11N	Cloudy
7–8	32.0	41.0	208.6	155.0	53.6	20,520	1,099,870	13N	Cloudy
8–9	32.0	41.0	205.0	153.0	52.0	18,960	985,920	13N	Cloudy
9–10	31.8	41.2	202.0	149.0	53.0	18,720	992,160	14N	Cloudy
10–11	31.9	41.1	200.3	147.0	53.3	18,780	1,000,970	15N	Cloudy
11–N	32.4	40.6	200.0	146.0	54.0	18,540	1,001,160	11N	Cloudy
P. M.									
N–1	32.8	40.2	201.0	146.5	55.5	18,780	1,042,900	12N	Cloudy
1–2	32.8	40.2	202.0	147.8	54.2	18,600	1,008,120	12N	Cloudy
2–3	32.4	40.6	204.2	148.4	55.8	18,600	1,037,880	11N	Cloudy
3–4	32.0	41.0	207.0	150.2	56.8	18,300	1,039,400	9N	Cloudy
4–5	33.0	40.0	205.0	151.0	54.0	18,600	1,004,400	7N	Cloudy
5–6	33.5	39.5	204.2	149.3	54.9	18,420	1,011,260	9N	Cloudy
6–7	33.3	39.7	207.8	150.5	57.3	18,300	1,048,590	5N	Cloudy
7–8	33.0	40.0	206.5	151.3	55.2	18,300	1,010,160	6N	Cloudy
8–9	33.1	39.9	209.0	151.2	57.7	18,600	1,073,220	6N	Cloudy
9–10	33.1	39.9	210.6	153.0	57.6	18,600	1,071,360	4N	Cloudy
10–11	33.2	39.8	210.4	154.0	56.4	18,600	1,049,040	3N	Cloudy
11–M	33.0	40.0	209.0	154.0	55.0	18,840	1,036,200	3N	Cloudy
Total	950.7	x 31,750 =	30,184,700 Btu				Total Btu supplied = 24,818,090	Average 10.2 Mph	

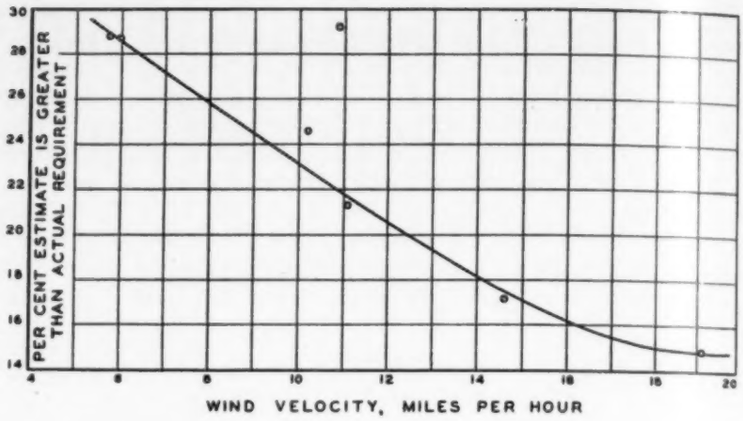

Fig. 3—Curve showing per cent estimated heat requirements are greater than actual plotted in relation to wind velocity

The estimated and the actual heat requirements were determined in the following manner. From the average hourly indoor temperature, secured from the 40 thermometers distributed throughout the building, the average hourly outdoor temperature was subtracted. This difference, multiplied by 31,750 Btu, the estimated heat requirement per degree-hour, gave the estimated hourly heat requirement. The hourly heat requirements were then totaled for each 24-hour period.

From the flow meter records the number of pounds of water flowing through the heating system each hour was determined and this, multiplied by the average hourly difference in the temperature of the water in the flow main and of that in the return main, for the same hour, gave the quantity of heat delivered to the building during that hour. The hourly quantities of heat delivered were then totaled for each 24-hour period.

The observations reported in this paper were all made on totally cloudy days. In this manner the effect of solar radiation was eliminated from the calculations. Weather conditions during these tests were as near design conditions as could be secured: that is, an outdoor temperature of 25 F and a wind velocity from the north of approximately 15 mph. There was no precipitation in any of the cases reported, except a very light snow on the morning of February 4.

On days when the sun was shining, the effect was immediately discernible in the rooms exposed to the sun. The temperature in those rooms rose rapidly to as much as 10 deg above that when the sun was not shining, while the temperature in the rooms on the shaded portion of the building remained normal. This clearly indicates that automatically controlled radiator circuits, zoned for the various exposures of a building, are very desirable.

After these studies had been made and the per cent of variation between the estimated and the actual heat requirements of the building found to be approximately 23 per cent, an additional check was made of the calculations used to estimate the heat requirements of the building. With a thickness gage, the cracks between the steel sash and the window frame were measured on a great number of windows of the building and found to be considerably smaller than those assumed. Consequently, the infiltration of air around the windows was corrected to more nearly conform with the actual crack found to exist for the windows in the building. This correction reduced the total estimated heat requirement under design conditions to 1,271,700 Btu per hour, or 28,900 Btu per degree difference, indoor to outdoor temperature. Thus, the estimated heat requirements were reduced to 13.5, 18.3, 17.7, 10.5, 4.6, 17.3, and 6.5 per cent higher than the actual heat requirement, or an average of 12.6 per cent more estimated than required.

In Fig. 3 the average wind velocity per hour for each of the seven 24-hour periods reported was plotted against the percentage the estimated heat requirements were higher than the actual heat consumed. Although more data are desirable, a definite trend is indicated in the relation between wind velocity and percentage of variation between the estimated and the actual heat requirements of the building.

From this study it appears that the greatest source of error in making an estimate of the heat requirements of a building occurs in allowances made for air infiltration. A calculation shows that if the air change method, based upon Table 3, Chapter 6, of THE 1935 GUIDE, were followed, even allowing only one air change per hour, the heat requirement estimate would be considerably larger than indicated above.

Professor Diederichs Elected Dean of Cornell College of Engineering; Professor Hollister Associate Dean

Announcement has been made by Pres. Livingston Farrand of Cornell University, Ithaca, N. Y., of the election by the Board of Trustees of Prof. Herman Diederichs, director of the Sibley School of Mechanical Engineering and the holder of the John Edson Sweet Professorship of Engineering, as dean of the College of Engineering at Cornell University. Professor Diederichs will succeed Dean Dexter S. Kimball, who will retire on July 1.

At the same time the election was made known of Prof. S. C. Hollister as associate dean of the College of Engineering, effective July 1, in addition to his regular duties as director of the College of Civil Engineering.

Professor Diederichs is regarded as an authority in experimental engineering and he is the author of text books and other publications. He is also the co-author of several bulletins of the Cornell Engineering Experiment Station among which are the Heat Transfer from Steam to Heavy Fuel Oil, and Flame Propagation in Closed Cylinders.

The Infiltration Problem of Multiple Entrances

By **A. M. Simpson,* New York, N. Y.** *and* **K. B. Atkinson**, New Brunswick, N. J.**
(MEMBERS)

FOR many years air conditioning engineers have discussed the effect of closed doors and windows in buildings cooled for comfort, and the point has been stressed that cooling outside air is quite expensive so it is desirable to reduce as far as possible various sources of infiltration.

In buildings where there are entrances in more than one outside wall located so that wind pressures may produce a through draft across a floor, mere reliance on door checks to keep the doors closed and exclude infiltration fails to solve the problem. In cities such as New York and Buffalo where summer wind velocities average over 12 mph, and in Chicago where they average over 10 mph, the pressure differential built up across a building produces air velocities through doorways frequently as high as 5 mph. Drafts traveling at this speed sweep through a building from one entrance to the other every time the doors are simultaneously opened, and obviously the greater the traffic and activity of business the worse the resulting conditions.

All air brought in from the outside is a cooling load, but some air must be admitted for ventilation. Air, in excess of ventilation requirements, if allowed to enter, is a needless cooling load. Furthermore, air for ventilation should enter through the cooling apparatus so that it may be properly cleaned, controlled, distributed so as to reduce drafts, and so that the capacity of the apparatus will be decreased to satisfy the load conditions.

When there are high winds entrance infiltration may greatly exceed the air requirements needed for ventilation and under such conditions if door openings are depended upon for ventilation, all control may be sacrificed. Moreover, air introduced in this manner is by far the least desirable method of allowing for the proper amount of ventilation. Air swept across the scorching pavements into an air conditioned enclosure may only be cooled by the other air in the buildings which must necessarily be sub-cooled to produce satisfactory results. Also the magnitude of this localized load greatly adds to distribution difficulties. Air coming in through entrance doors is laden with dirt from the streets and can neither be washed nor filtered, thus offsetting one of the principal benefits of summer air conditioning, that of the elimination of dust and dirt carried by drafts and summer winds, with a corresponding reduction in the depreciation and spoilage of merchandise on display. Outside air brought through the apparatus to meet ventilation requirements may be drawn from cooler, cleaner, shady areas; and it can be washed or filtered; all of which tends to decrease the capacity of the equipment, thus lowering original cost and increasing efficiency. Any air infiltrated in excess of ventilation requirements is simply a needless load necessitating added equipment and capacity and increasing operating costs.

The investigation of entrance infiltration seems to have been somewhat overlooked by the heating and ventilating engineer. Up until the present time no coefficients or rules for estimating these losses have been worked out and presented to the profession. While there have been accurately determined and tested heat transmission coefficients by which conduction, convection and radiation losses can be closely estimated under any given condition, estimating infiltration or exfiltration losses through entrance doors has so far been a matter of rough approximation. The lack of experimental data and established coefficients is undoubtedly due to the fact that the problem of entrance infiltration is not one that lends itself to laboratory experiment or determination. It is a problem that may only be studied under the widely varying conditions that occur in the field.

Sources of Entrance Infiltration

There are three distinct phases of the problem of entrance infiltration which may be stated as follows:

1. Entrance infiltration caused by combinations of wind pressure, chimney effect of heated building and exhaust fan ventilation in the heating season.
2. Infiltration and exfiltration caused by wind pressure acting on two or more entrances in different outside walls around a given area in which temperature and low humidity are to be maintained in the comfort zone during the warm weather season.
3. Infiltration and exfiltration caused by wind pressure and temperature difference acting on a single entrance to an air conditioned space.

From the point of view of the most popular form of air conditioning, comfort cooling, the second phase of the problem, losses due to multiple entrances facing different streets, is the most serious, and this discussion will be confined to that part of the problem. The data compiled from extensive field measurements of infiltration and exfiltration losses occurring in buildings or rooms having doors on more than one outside wall are presented as a basis for establishing more accurate estimating practice in this field.

Effect of Prevailing Wind and Wind Pressure

The losses through multiple entrances in mechanically cooled buildings are caused largely by differences in wind pressure. However, one of the first rather surprising facts developed by field tests is that the direc-

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Table 1—Wind Velocities Under Conditions of Typical Design Wet-Bulb Temperatures[a]

	Month	Day	Hour of Day—Standard Time 9 A.M.	10	11	12	1 P.M.	2	3	4	5	6
Wet-Bulb	June	6	72	73	73	74	74	74	74	75	73	70
Wind Velocity			12	12	12	13	12	13	12	15	15	
Wet-Bulb	June	21	71	73	74	74	74	74	74	74	73	73
Wind Velocity			11	12	12	15	15	16	18	17	14	
Wet-Bulb	June	24	71	72	72	73	72	75	72	72	73	73
Wind Velocity			6	8	9	12	10	12	12	12	12	
Wet-Bulb	June	29	79	79	80	82	82	82	79	77	75	73
Wind Velocity			11	11	10	11	10	12	13	14	10	
Wet-Bulb	June	30	69	71	72	71	73	74	75	75	75	75
Wind Velocity			3	5	7	6	7	7	8	7	7	
Wet-Bulb	July	3	74	74	76	75	75	75	76	74	76	75
Wind Velocity			10	9	11	12	12	11	11	10	11	
Wet-Bulb	July	6	76	77	77	78	79	76	78	79	79	77
Wind Velocity			7	7	8	10	11	12	9	11	12	
Wet-Bulb	July	7	81	79	79	77	76	78	76	77	76	77
Wind Velocity			10	11	10	12	12	12	11	13	9	
Wet-Bulb	July	14	71	72	73	74	77	75	75	75	76	74
Wind Velocity			8	9	8	9	7	7	7	7	7	
Wet-Bulb	July	15	72	74	75	76	77	78	77	76	75	72
Wind Velocity			5	8	10	11	12	12	11	10	8	
Wet-Bulb	July	20	77	78	78	78	79	79	73	73	71	73
Wind Velocity			9	9	10	9	8	14	15	9	9	
Wet-Bulb	July	21	76	77	78	75	76	77	74	74	73	72
Wind Velocity			4	6	8	11	9	11	13	11	10	
Wet-Bulb	July	25	68	69	72	74	76	78	80	79	78	78
Wind Velocity			13	12	9	11	10	9	9	11	11	
Wet-Bulb	July	26	78	77	79	79	81	80	80	79	77	78
Wind Velocity			3	6	7	8	9	9	10	11	11	
Wet-Bulb	July	27	77	77	76	77	76	77	78	78	80	78
Wind Velocity			9	8	9	13	12	12	12	12	12	
Wet-Bulb	July	28	78	77	77	76	71	73	74	76	77	76
Wind Velocity			9	10	9	9	5	6	7	7	5	
Wet-Bulb	August	10	75	76	76	76	76	75	76	76	76	77
Wind Velocity			10	10	10	10	11	10	11	10	9	
Wet-Bulb	August	11	75	77	74	75	75	75	72	74	74	72
Wind Velocity			2	3	2	5	7	8	9	9	8	
Wet-Bulb	August	24	70	70	72	74	77	76	77	76	75	76
Wind Velocity			15	12	10	10	11	12	12	11	12	
Wet-Bulb	Sept.	4	68	71	72	74	75	76	76	75	73	72
Wind Velocity			10	11	10	12	12	12	11	9	10	

Ave. wind velocity during periods of 75F wet-bulb or higher, 9.7 mph.
Ave. wind velocity of wind throughout four months, 8.8 mph.
[a]From 1934 report of New York Meteorological Observatory of the Department of Parks, Central Park, New York, N. Y., United States Weather Bureau.

tion of the prevailing wind has little effect on the direction of the air passage through an entrance under the usual conditions of exposure in any of our large cities. The obstruction and interference with the air currents at the street level brought about by surrounding buildings of varying heights causes the forces of wind velocity to be transformed into eddy currents and areas of varying pressures. These create entrance velocities in surprising directions, and even in some cases supposedly protected entrances may prove to be serious problems.

As an example: a small restaurant on the south side of 37th Street in New York City with an entrance facing north was checked on a day when there was a southwest wind blowing 18 mph. This restaurant which has a front and a rear doorway showed an infiltration velocity at the north entrance of 3 mph and an exfiltration velocity at the south entrance of 2½ mph. A high building facing south on the north side of 37th Street directly opposite showed an exfiltration velocity of 1¾ mph through the entrance facing south. The explanation is that the southwest wind piling up against the high building on the north side of the street caused eddy currents and static pressure which reversed the draft through the restaurant. Also the wind blowing into open windows on the upper floors caused a pressure inside the tall building which resulted in a reversed infiltration at the street level.

This example is typical of the reactions that may be expected in congested centers where the obstruction of buildings causes eddy currents and pressure areas. In the few typical entrance tests given in Table 2 several instances are shown where infiltration was in the direction opposite to or partially opposite to the direction of the wind velocity at the time the test was made. Where a structure is directly opposed to the prevailing wind without obstruction, as would be expected, the infiltration velocity is found to follow approximately the formula:

$$V_1 = 0.6V$$

where: V_1=velocity through aperture, fpm
V=component of wind velocity at right angles to doorway, fpm
0.6—orifice coefficient of the aperture

Wind Velocities Occurring with Design Wet-Bulb Temperatures

A theory current with many air conditioning engineers is that when design wet-bulb temperatures of 74 F and above, for example, occur simultaneously with bright sunshine, the wind velocity is likely to be low. Therefore in sizing the cooling apparatus for maximum design wet-bulb temperature and solar radiation, the capacity for this infiltration frequently is not considered.

A study of the U. S. Weather Bureau record shows the fallacy of this conception. Table 1 compiled from records of the New York Meteorological Observatory gives hourly wet-bulb temperatures and wind velocities for every day during 1934 when the wet-bulb temperature in New York City reached 75 F or higher. Examination of the table makes it apparent that the highest wind velocities frequently occur when the wet-bulb temperature is also at a peak. As a matter of fact the average wind velocity during the periods of 75 F wet-bulb or higher is 9.7 mph, whereas the average wind velocity throughout the four months of June, July, August and September is 8.8 mph.

Method of Determining Infiltration and Exfiltration Velocities

The development of the procedure for field testing to determine the velocity of infiltration or exfiltration through an entrance under varying conditions resulted in a return to basic principles. Anemometers were found to be unreliable with the varying and intermittent velocities encountered. The Kata thermometer was not adapted to effective use in a field test conducted in the busy entrance to a store or office building.

It was found that the most accurate and reliable measurement of infiltration and exfiltration velocities resulted from the use of a smoke puff test. This was conducted by timing the travel of puffs of smoke over a measured distance. The smoke puffs were emitted throughout a traverse of the entrance and the average velocity over the measured distance was taken as the velocity of infiltration or exfiltration under the prevailing conditions.

Since these tests were conducted at busy entrances under actual field conditions it was necessary to have a means of producing puffs of smoke that would not be

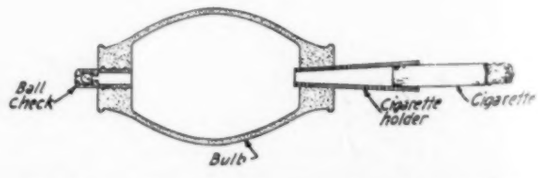


Fig. 1—Smoke puffer for measuring air movement

objectionable to the public, and which could be made rapidly at frequent intervals in order to cover a traverse of the doorway in a brief period of time so as not to interfere with the flow of traffic. To accomplish this result the device illustrated in Fig. 1 was developed to provide puffs of smoke from an ordinary cigarette.

The test was conducted by three men. First a convenient distance was measured off from the doorway toward the interior of the store and a similar distance toward the exterior. Men were stationed at the interior and exterior measured distances with stop watches. The man at the doorway, using the smoke puffer, emitted puffs of smoke over a traverse of the entrance using at least five points, the two upper quarters of the doorway, the two lower quarters of the doorway and the center of the doorway. The men stationed at the measured distance from the doorway timed the period of travel for the smoke from the doorway to the line which they observed. With the distance known, the velocity of travel and direction either for infiltration or exfiltration could be observed. In each case the dry-bulb temperature, the wet-bulb temperature, the wind velocity and wind direction were obtained from the local Weather Bureau records for the time of the test.

Of the 187 tests conducted to determine the maximum and minimum conditions to be expected and the mean or average infiltration or exfiltration velocity in buildings and rooms having doors in more than one outside wall, the results indicated in Table 2 are typical. A summary of the values tabulated shows that infiltration and exfiltration velocities from areas having multiple exposed entrances will vary normally from a maximum of 350 fpm to a minimum of 104 fpm with a mean of 256 fpm.

Cost Analysis of Open Doorways.

The results obtained for the velocity of infiltration and exfiltration drafts in the case of multiple exposed entrances indicate the impracticability of maintaining conditions in buildings where cross drafts through open doorways are possible. In the case of two entrances both remaining open and a mean infiltration and exfiltration velocity of 256 fpm, assuming a double doorway 5 ft wide by 7 ft high, the volume of infiltration and exfiltration would amount to 9000 cfm. With a 12 F differential of cooling this additional load would demand a refrigeration capacity of 32 tons. However, even though this additional tonnage were supplied, there would be no control of dirt and moisture content, and excessive sub-cooling and dehumidifying of the processed air would be required to absorb the load and maintain satisfactory average conditions. Moreover, entrance arrangements of this type impose serious air distribution problems.

Assuming ventilation requirements for 200 persons, which normally would be supplied through the apparatus, as a credit to the infiltration loss, there would still be 5700 cfm of unconditioned air which would have to be cooled and dehumidified by contact and mixture with sub-cooled and dehumidified air from the refrigerating apparatus. With a 12 F differential of cooling this additional amount of air would require a cooling capacity of 15½ tons. It is therefore evident that it is impracticable to attempt efficient or effective cooling and air conditioning when two or more opposite or cross draft entrances are standing open.

Tests on the Swing Door Entrances

Any swing door or combination of swing doors, including vestibules, constitutes an open or periodically open orifice through which air is passed in or out of a building. The general laws of an orifice therefore apply and the quantity of air flowing is equal to the area of the doorway multiplied by the duration of the open period and the velocity of air through the doorway. Since the velocity was determined by averaging the velocity read-

Table 2—Entrance Draft Velocities in Multiple Entrance Buildings Under Summer Cooling Conditions

Building	Door Location	Velocity fpm	Meteorological Conditions			
			Dry-Bulb F	Wet-Bulb F	Wind Vel. mph	Wind Direction
Christine Tea Room 18 E. 37th St., New York	N S	+ 361 − 350	85	70	12	SW
Rogers Peet 42 St. and 5th Ave., New York	N SW	− 480 + 370	83.4	66	18	SW
Lofts 42 St., New York	S N	+ 500 − 280	83.4	66	18	SW
Woolworth's B'way at 34 St., New York	E S	+ 300 − 246	77	66	25	S
Macy's, 7th Ave., New York	W	+ 120	77	66	25	S
Horn and Hardart 7th Ave. and 34 St., New York	W	+ 267	77	66	25	S
Hotel New Yorker 34 St., New York	S	+ 560	77	66	25	S
Ovington's 5th Ave. and 39 St., New York	W N	− 226 + 327	82	62	8	W
Emigrant Savings 42nd St., New York	S N	+ 287 − 321	82	62	8	W
Hotel Bradford Tremont Street, Boston	E	− 290	88	70	12	SW
Hofbrau Restaurant Tremont St., Boston	E	+ 176	88	70	12	SW
Royal Cafe and Grill Tremont St., Boston	W	+ 180	88	70	12	SW
Hotel Touraine, Boston Lobby Entr., Tremont St. Lobby Entr., Boylston St.	W N	+ 120 + 264	88	70	12	SW
Peter Flynn Co. Tremont St., Boston	W	+ 136	89	71	12	SW
Jackson and Co. Tremont St., Boston	W	+ 130	89	71	12	SW
Chandler and Co., Boston West St. Tremont St.	N W	− 210 + 102	89	71	12	SW
Jordan Marsh, Boston Cor. Entr. Washington and Temple Washington St. Entr. Summer St. Entr. Temple St. Entr. Annex Temple St. Entr. Bedford St. Entr.	SW W N S N S	+ 384 − 140 + 128 − 140 + 144 − 148	89	71	12	SW
Parker House, Boston Tremont St. Entr. School St. Entr.	W N	+ 170 − 275	89	71	12	SW
Rogers Peet Co., Boston Tremont St. Entr.	W	− 384	86	70	12	SW
Malley's Dept. Store, New Haven, Conn. 904 Chapel St. Temple St.	N W	− 169 + 188	85	69	8	S
Woolworth's New Haven, Conn.	N W	+ 190 − 162	85	69	8	S
Lord Baltimore Hotel Baltimore, Md.	S W	+ 158 − 130	90 90	75 75	8 8	NE NE
Read Drug Co. Baltimore and Hanover Baltimore, Md.	S S	+ 300 + 300	90 90	75 75	8 8	NE NE

ings from a traverse of the door area, the customary reduction coefficient used with velocity of approach need not be used. However it is desirable to consider the frequency and extent of through opening at which the velocity of the air determined previously may sweep through a multiple entrance.

Duration of Door Opening

Tests were conducted in over 150 building entrances in cities from New York to Chicago and from Atlanta to Boston to determine the duration of the effective door opening for each person passing through hinged doors. These tests were conducted by clocking the traffic in and out of these several entrances and noting with a stop watch timer the duration of the effective door opening during several periods of 100 or more operations. The watch was started each time the door started to open and was allowed to run as long as the door remained open and until it closed to a point within 12 in. of the jamb, so as not to give a false reading due to the delayed final action of the door check. The operation of the stop watch timer gave the accumulated duration of effective opening for the passages counted during each test. These tests indicated the effective door opening for each person passing through the entrance was an average of two seconds per person passing through a single set of swing doors. The data from a few typical tests are shown in Table 3. It was found that the average swing door required 2 seconds to open and approximately 5 seconds to close from the full open position to the tight closed position.

Table 3—Tests of Single Swing Door Entrances and Swing Door Vestibule Entrances to Determine the Duration of Effective Door Opening for Each Person Passing Through the Entrance

Location New York City	Duration of Test in Min.	Traffic Through Building No. People			Total Time in Seconds Inner and Outer Doors of Vestibule Open Permitting Through Draft	Duration in Seconds of Draft Through Opening per Person
		In	Out	Total		
Duration of Draft Through Single Swing Door						
R. H. Macy and Co. 7th Ave. Entr.	5	86	76	162	290	1.80
Littmann's B'way and 38 St.	5	48	51	99	202	2.04
Schrafft's B'way and 37 St.	5	12	13	25	59	2.36
					Average	2.06
Duration of Draft Through Vestibule						
R. H. Macy and Co.	5	120	100	220	315	1.43
Arnold Constable 5th Ave. and 40 St.	5	72	74	146	226	1.55
Paramount Bldg.	5	77	62	139	212	1.52
					Average	1.50

Tests observed on a large number of buildings having entrances in more than one outside wall under conditions suitable for summer cooling show an average infiltration or exfiltration velocity through the doorway of 256 fpm. Assuming the area of the average swing door to be 20 sq ft, the average infiltration per person through a single set of swing doors having entrances in more than one outside wall is:

$$2 \text{ seconds draft opening per person} \times 20 \text{ sq ft} \times \frac{256 \text{ fpm}}{60} =$$

171 cu ft per person.

Assuming that a small dollar chain store having entrances facing on two streets has an average of 600 entrance passages per hour at 171 cu ft per person in and out the draft through the entrances would produce an infiltration of 12,000 cu ft of outdoor air per hour. With a 12 F differential of cooling this infiltration would represent a cooling load of approximately 4 tons of additional refrigeration capacity. The unknown heat gain from infiltration is one of the reasons temperature and humidity guarantees for air conditioning applications are frequently qualified where there are entrances in more than one outside wall, unless the doors are permanently closed on one side.

Tests on Vestibules for Multiple Entrances

In overcoming the difficulties produced by multiple entrances the vestibule presents itself as a possible solution. However, an investigation of their effectiveness indicates the reason vestibule entrances do not materially assist in maintaining guaranteed conditions. With heavy traffic through multiple entrances, both the inner and outer doors are open at the same time so frequently as to provide a duration of a through draft opening for each person entering, which is apparently not much better than a single door. This is an obvious difficulty with shallow vestibules and attempts are frequently made to overcome this condition by using deep vestibules. However, it is rarely practicable to arrange the distance between the two sets of doors more than 8 to 10 ft apart.

Actual tests show a vestibule arrangement to be less satisfactory than is apparent to the casual observer. Tests were conducted on 200 vestibule entrances over the same broad range of territory in which tests were conducted on the single swing door entrances. These tests were made in order to determine the exact effectiveness of arrangements of vestibule entrances using swing doors and to determine the effective through-door opening in a vestibule for each person passing through the entrance. A similar procedure was followed as in the case of the tests for the single swing doors. The traffic through each entrance was clocked in and out through the vestibule and the duration of the through draft opening was timed, when at least one inner and outer door was opened. The watch was stopped when either the inner or outer set of doors was closed to within 12 in. of the jamb. A few typical tests showing average results are given in Table 3.

It was determined that in busy entrances the effective duration of a through draft opening was 1.5 seconds for each person passing through the entrance which indicated that this type of vestibule gives only 25 per cent better protection than a pair of swing doors. The failure of the vestibule to give better protection comes from the fact that the flow of usual traffic is such that the repeated simultaneous opening of the inner and outer doors nullifies the expected protection.

It will be noted that where the normal infiltration per person passing through a single swing door into a building with entrances on more than one outside wall, would

be 171 cu ft per person, the use of a vestibule would reduce this infiltration to 128 cu ft per person. Referring to the example of the small dollar chain store with single swing door entrances and where the infiltration losses required 4 tons additional refrigeration capacity, the use of vestibule entrances instead of single swing door entrances would still allow infiltration and exfiltration losses that would represent a cooling load of 3 tons additional refrigeration.

Results With Revolving Doors

The use of revolving doors as a solution to the multiple entrance problem in connection with air conditioning has been considered rather extensively due to the fact that the revolving door acts as an air lock and will prevent air sweeping through an entrance to produce drafts. It has generally been considered that this type of door was a form of metering device and engineers have felt that every revolution of a revolving door forced an exchange of inside and outside air which would not occur with swing door entrances, except under most unfavorable conditions.

The factors which govern the displacement of air by a revolving door are:

1. Speed of rotation of the revolving door.
2. Scavaging effect of air currents directed toward the compartments of the door when they are exposed at the interior and exterior of the entrance.
3. The difference in temperature between the interior and exterior, which would set up a vertical circulation of the air in each compartment as it was exposed to the interior and to the exterior.

Since a revolving door seals the entrance against any free flow of air, the size of the room, the height of the building and the difference in pressure between the inside and outside has no measurable or practical effect on the displacement of air produced by the revolving door.

Apparatus was arranged to determine the efficiency of a revolving door as a metering device, and to establish data on the cubic feet of infiltration per person passing through an opening of this design at various traffic densities. In order to simulate conditions as severe or more severe than conditions in actual field service and also to provide a means of measuring the displacement of air by a revolving door and its efficiency as a metering device, an apparatus was arranged as shown in Fig. 2. A room 8 ft square and 8 ft high was built of insulating material and at the entrance to this room a revolving door 7 ft in diameter by 7 ft high was installed in the usual manner of applying a revolving door to an entrance. A 15 in. ventilating fan was located at one corner, directed approximately as shown in Fig. 2, to provide a scavaging velocity of not less than 100 fpm under all test conditions. An ice bunker was built inside of the room to provide a means of maintaining a difference in temperature between the interior and exterior of the booth when the revolving door was rotated at a constant speed.

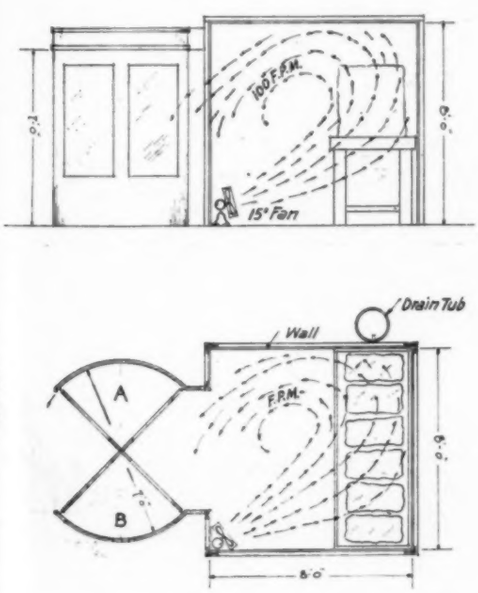


Fig. 2—Elevation and plan of test room for determining efficiency of revolving door

Referring to Fig. 2 it is evident that if the revolving door were 100 per cent efficient as a metering device the temperature in the quadrant *A* would always be the same as the temperature in the interior of the booth and the temperature in quadrant *B* would always be the same as at the exterior of the revolving door. As the temperatures at *A* and *B* approached equality there would be an indication that the air in the compartments of the revolving door was practically dead air rotating with the revolving door and was being tempered equally by the air from the interior and exterior. Thermometers were placed in the interior and exterior of the booth and also in compartments *A* and *B*. In each test that was conducted it was found that the temperatures in *A* and *B* were as nearly identical as it was possible to read them, on thermometers calibrated to 0.1 F. This indicated that as a metering device the revolving door was far from 100 per cent efficient; in fact, that its efficiency must be very low.

In order to secure an accurate measurement of the amount of infiltration and exfiltration caused by a revolving door, a test was conducted to measure the heat loss of the melting of ice within the test room caused by the rotation of the revolving door at varying speeds. First, the heat transmission factor for the complete booth, revolving door, walls and ceiling was estimated on a stationary basis and checked by actual measurement of the ice melted over a given period. During the test the fan was operated to give a uniform temperature throughout the room and readings were observed on 7 thermometers in the booth and 6 thermometers outside of the room to determine the temperature difference.

After the heat transmission factor was determined for the room with the revolving door stationary, the revolving door was then operated at various speeds under similar conditions. With thermometers arranged as in the stationary test, dry- and wet-bulb temperatures were taken at frequent intervals on the interior and exterior of the booth with the fan operating. The amount of ice melted was weighed in a series of tests in order to determine the difference in the amount of ice melted with the revolving door in operation and by this method the amount of infiltration and exfiltration caused by the revolving door was calculated.

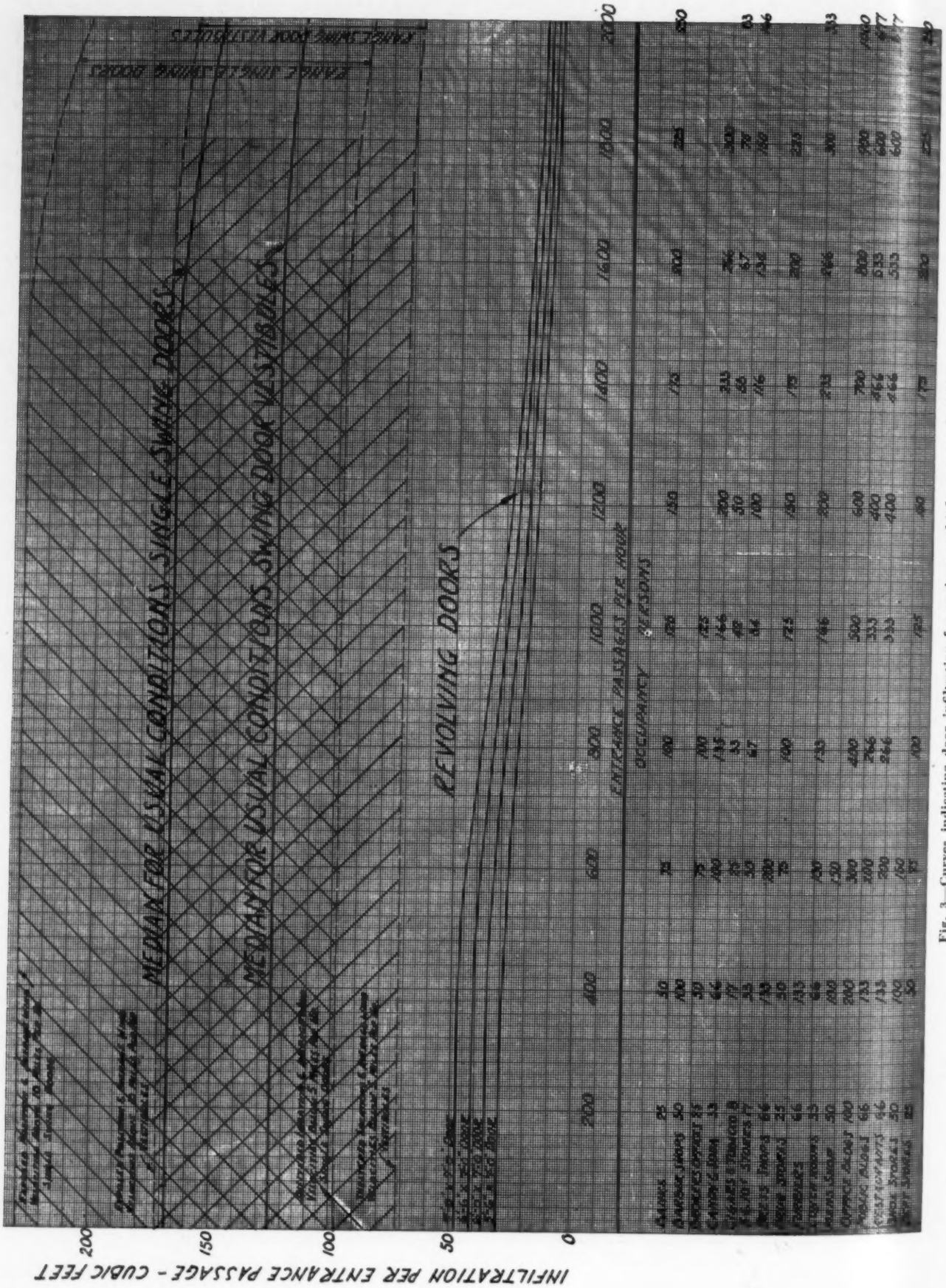


Fig. 3—Curves indicating door infiltration for various multiple entrance buildings

The results showed that under a normal operating speed of approximately 13 rpm the revolving door was found to be approximately 17 per cent efficient when considered as a metering device. The amount of infiltration and exfiltration caused by each revolution of this size revolving door was approximately 45 cu ft per complete revolution. This method was further checked by replacing the ice bunker and ice supply with a steam radiator. It was felt that this would simplify the test by eliminating any possible variations due to slight difference in wet-bulb readings. A similar test procedure was adopted in that the heat transmission factor of the test room was first determined with the revolving door stationary and then a test was conducted with the revolving door in operation and measuring the increase in weight of the condensate. It was found that with the revolving door operating at the same speed as in the previous test, the result was approximately the same as in the test when using the ice bunker.

One further test was made to obtain an additional check on the efficiency of the revolving door as a metering device. This was accomplished by the use of a photo-electric cell and sensitive ammeter with the electric eye connected across the booth. In various tests the location was varied from side to side, from front to back and from top to bottom. During the test the 15 in. propeller fan was operated in the booth to insure turbulence that would give uniform density of smoke throughout the booth and this fan was continued in the same direction of discharge so as to provide a scavaging action on the compartments of the revolving door as they rotated. Under these conditions a smoke candle was set off in the booth. The revolving door was rotated at a normal speed of 12 rpm and readings were taken on the gage as soon as a stable condition was indicated within the range of accuracy for the instrument. The readings were plotted on a curve to indicate the relative dilution of the air in the booth due to the rotation of the revolving door.

The formula for the percentage of the volume displaced after any given number of revolutions is:

$$V=(1-D)^R$$

Where V=Percentage of volume of booth displaced
D=Per cent displacement of volume per revolution of the revolving door.
R=Door revolutions.

Applying this formula to the curve it was determined that 9 per cent of the test room's 512 cu ft was displaced by each revolution of the revolving door. 9 per cent × 512 cu ft = 46 cu ft per complete revolution, which was an additional check on the previous tests based on heat transfer.

It was observed that the displacement of air by a revolving door depended upon the velocity of rotation but since the velocity of the rotation of a revolving door under normal operating conditions is determined by the rate of walking, analysis showed that within the limits of this variable the revolving door would not be more than 20 per cent efficient as a metering device.

Another factor which was noted in determining the amount of infiltration by a revolving door was the number of revolutions required for the passage of each person through a revolving door. A study of this factor was made in over 180 revolving door installations, including revolving doors of all ages and conditions of repair and maintenance. The number of people passing through the revolving door in each case was clocked over a considerable period of time and the number of revolutions of the revolving door were counted at the same time. With new revolving doors equipped with speed governors, the revolutions for each person passing through varied between 0.20 and 0.40 revolutions per person. With older revolving doors in a poorer state of repair the variation was between 0.35 and 0.53 revolutions per person passing through. It is generally safe to assume that a revolving door which is fairly well maintained will average not over 0.40 revolutions per person.

The curves shown in Fig. 3 are for various sizes of revolving doors giving the infiltration per entrance passage, plotted against entrance passages per hour as determined from the tests. Above the curves showing revolving door infiltration are corresponding curves for single swing door entrances and vestibule entrances where they are in use in buildings or rooms having entrances on more than one outside wall.

Conclusions

While infiltration in individual multiple entrance installations varies considerably, depending on prevailing wind velocity and direction, and upon the location of surrounding buildings, a mean draft velocity of 256 fpm may be used where swing doors are used. The range of values for infiltration of single sets of swing doors is 100 to 230 cu ft per entrance passage depending on conditions of exposure, prevailing wind velocity, etc., and 72 to 170 cu ft per entrance passage for vestibules. The infiltration per entrance passage is 10 to 50 cu ft for a revolving door depending on the size of the door, on the building occupancy, and on the corresponding entrance passages per hour. High winds will not materially change these values since there is no draft opening involved, as the door acts as a seal and the air which is passed depends upon its limited action as a metering device.

The investigation of entrances in general and multiple entrances or entrances to rooms having doors on more than one outside wall indicates the following results:

1. Average duration of draft opening through single set of swing doors per entrance passage is 2 seconds.

2. Average duration of draft opening through a vestibule per entrance passage is 1.5 seconds.

3. Under conditions when air cooling would be in operation the mean draft velocity where there are opposed multiple entrances is 256 fpm.

4. Under conditions when air cooling would be in operation in a room having doors in more than one wall the infiltration per entrance passage through a single set of swing doors is 100 to 250 cu ft, with an average of 171 cu ft.

5. Under conditions when air cooling would be in operation in a room having doors in more than one wall the infiltration per entrance passage through a vestibule is 72 to 170 cu ft, with an average of 128 cu ft.

6. Infiltration per entrance passage through a revolving door is 10 to 50 cu ft depending on the size of door and intensity of traffic.

W. P. Culbert
Golf

M. F. Blankin
Banquet

J. H. Hucker
Entertainment

PHILADELPHIA CHAPTER

W. F. Smith
Vice-Chairman

R. C. Bolsinger
General Chairman

H. H. Erickson
Publicity and Transportation

COMMITTEE ON ARRANGEMENTS

H. G. Black
Registration

L. P. Hynes
Finance

M. C. Gillett
Ladies

THE POCONOS IN JUNE

A.S.H.V.E.....Buck Hill Falls

A.S.R.E.....Sky Top

SOON members of the A.S.H.V.E. and their guests will meet at Buck Hill Falls, Pa., for the Society's Semi-Annual Meeting, June 22-23-24, where they will again enjoy the hospitality of The Inn and the plans made for their entertainment and pleasure by the Committee on Arrangements of the Philadelphia Chapter.

The accompanying program shows the technical papers which will be presented and the social features. An important event will be the joint session with *A.S.R.E.* members at The Inn on Tuesday morning, June 23. On this occasion W. H. Carrier, Newark, N. J., a past president of both societies, will preside and after the technical session adjourns the Councils of the A.S.H.V.E. and the *A.S.R.E.* will have a joint luncheon.

Several committee meetings have been scheduled and members of the Committee on Research will have a luncheon at 12:15 p. m. on June 22. On Tuesday, June 23, the Nominating Committee will meet at breakfast at 8:30 a. m., and a luncheon meeting of the Guide Publication Committee will be held at 12:15 p. m.

A variety of entertainment is offered. On Monday afternoon the golfers will play a Kicker's Handicap on the Buck Hill Course, while others will enjoy bowling on the green, tennis, or a trout fishing party. At 6:00 p. m. it will be time for members and ladies to leave for Skytop Mountain for a picnic and steak fry with *A.S.R.E.* members.

On Tuesday a men's golf tournament will be played for the Research Cup and the Philadelphia Chapter Cup. Those who do not play golf will have an opportunity to enjoy a sightseeing trip through the Poconos, horseback riding, or outdoor shuffle board. In the evening all members and ladies will attend the Semi-Annual Banquet and Dance, which will feature singing by the Adelphia Quartet, music by the Adelphia Orchestra, and group singing led by Christie, the accordionist.

Wednesday afternoon will provide an occasion for members to test their skill on the Skytop Golf Course with *A.S.R.E.* members, or to engage in trout fishing or lawn bowling.

The ladies will enjoy The Inn and the entertainment program which has been planned especially for them. M. C. Gillett, chairman of the Ladies Committee, will be assisted by Mrs. W. F. Smith, Mrs. M. C. Gillett, Mrs. R. C. Bolsinger, Mrs. M. F. Blankin, Mrs. L. P. Hynes, Mrs. H. G. Black, Mrs. J. H. Hucker, Mrs. H. H. Erickson and Mrs. W. P. Culbert. Miss Jane Louise Bolsinger will act as hostess.

In addition to the steak fry on Skytop Mountain and the banquet and dance, the ladies will meet for luncheon each day.

On Monday afternoon they will have a swimming party at the pool, followed by a reception and tea in honor of Mrs. G. L. Larson. On Tuesday morning they will have a choice of activities and can select a sightseeing trip through the Poconos, a golf tournament, or swimming. In the afternoon a luncheon and bridge party will be held at the Buck Hill Tennis Club. The program for Wednesday includes golf, lawn bowling, and swimming in the morning and outdoor games in the afternoon.

Those who plan to use railroad facilities to attend the Semi-Annual Meeting should purchase their ticket to Cresco, Pa., and inquire at the railroad office for the lowest available rate. Because of the reduced railway fares which will be in effect June 1, the Identification Certificate Plan has been cancelled and the customary identification certificates will not be distributed to members.

Situated in the Pocono Mountains, Buck Hill Falls provides delightful surroundings of natural scenic beauty and offers many varieties of outdoor recreation. Those who attended the Society's Semi-Annual Meeting two years ago will recall the laurel and rhododendron in bloom. The Committee on Arrangements extends a warm welcome, June 22-24.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annnual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

One of the Bridle Trails

Bowling on the Green

SCENES AT

One Shot to the Green

Looking Toward Skytop

BUCK HILL FALLS

A Good Catch

Where the Laurel and Rhododendron Grow

PROGRAM A.S.H.V.E. SEMI-ANNUAL MEETING

June 22-23-24—The Inn, Buck Hill Falls, Pa.

TECHNICAL

Monday, June 22, 1936

8:30 a. m. Registration

9:30 a. m. Greeting

Technical Papers:

Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg

Performance of an Underfeed Domestic Stoker, by T. G. Estep and D. C. Saylor

Relative Absorption of the Various Sections of a Round Heating Boiler, by A. J. Johnson and P. A. Mulcey

The Distribution of Steam in Heat Transfer Surface, by John McElgin

12:15 p. m. Luncheon Meeting, Committee on Research

Tuesday, June 23, 1936

8:30 a. m. Breakfast Meeting, Nominating Committee

9:30 a. m. JOINT SESSION of A.S.R.E. and A.S.H.V.E., W. H. Carrier presiding:

Theory and Practice of the Heat Pump, by C. W. Chamberlain

Application Factors Which Govern the Selection of Refrigerating Equipment for Air Conditioning Service, by J. R. Hertzler

Water Supply as Affected by the Demand for Summer Air Conditioning, by D. C. Morrow

The Quarter Century's Progress in Air Conditioning, by W. H. Carrier

12:15 p. m. Joint Luncheon—Councils of A.S.H.V.E. and *A.S.R.E.* (The Inn—Buck Hill Falls)

12:15 p. m. Luncheon Meeting, Guide Publication Committee

Wednesday, June 24, 1936

9:30 a. m. Technical Papers:

Development of Testing Apparatus for Thermostats, by D. D. Wile

Pyrheliometers and the Measurement of Total Solar Radiation, by L. A. Harding

A Field Study of the Heat Requirements of a College Building, by F. E. Giesecke and W. H. Badgett

Heating Requirements of an Office Building as Affected by Weather Conditions, by F. C. Houghten and Carl Gutberlet

ENTERTAINMENT

(For the Men)

Monday, June 22, 1936

1:30 p. m. Men's Golf Tournament—18 hole Kicker's Handicap for Members and Guests—Buck Hill Falls Course

Bowling on the Green

Tennis

Trout Fishing Party

6:00 p. m. Leave for Skytop Mountain

7:00 p. m. Picnic and Steak Fry on Skytop Mountain with *A.S.R.E.* Members.

Tuesday, June 23, 1936

1:30 p.m. Men's Golf—Buck Hill Falls Course—18 hole Medal Play Tournament for Members and Guests—Research Cup Tournament and Philadelphia Chapter Cup Tournament

Sightseeing Trip Through Poconos

Horseback Riding

Outdoor Shuffle Board

7:45 p. m. Banquet and Dance—Singing by Adelphia Quartet and Music by Adelphia Orchestra

Group Singing led by Christie—Accordionist

Wednesday, June 24, 1936

2:00 p.m. Optional Golf with Members of *A.S.R.E.* at Skytop Course

Trout Fishing Party

Bowling on the Green

(For the Ladies)

Monday, June 22, 1936

9:00 a. m. Registration

12:30 to 1:30 p. m. Luncheon—Main Dining Room, The Inn

2:00 p. m. Swimming Party at the Pool

4:00 p. m. Reception and Tea in Honor of Mrs. G. L. Larson

6:00 p. m. Leave for Skytop Mountain

7:00 p. m. Picnic and Steak Fry on Skytop Mountain with *A.S.R.E.* Members

Tuesday, June 23, 1936

9:00 a. m. Registration

10:00 a. m. Sightseeing Trip Through Poconos

Ladies' Golf Tournament—18 hole Medal Play—Buck Hill Falls Course

Swimming Party at the Pool

12:30 to 1:30 p. m. Luncheon and Bridge Party—Buck Hill Tennis Club

7:45 p. m. Banquet and Dance—The Inn

Wednesday, June 24, 1936

9:00 a. m. Ladies' Golf

Bowling on the Green

Swimming Party at the Pool

12:30 to 1:30 p. m. Luncheon—Main Dining Room—The Inn

2:00 p. m. Outdoor Games

Technical Program, 22nd SPRING MEETING A.S.R.E., Skytop Club, Skytop, Pa. June 22-23-24

Monday, June 22, 1936

10:00 a. m. Topic: Refrigeration of Foods

The Farmer's Egg Precooling Problem, by J. E. Nicholas

Defrosting, by S. Ruppricht

Application of Refrigeration to Brewery Stock Rooms, by Walter Jones

A Review of Fish Refrigeration Methods, by D. B. Finn

Tuesday, June 23, 1936

9:30 a. m. Joint Session with A.S.H.V.E. at The Inn, Buck Hill Falls, Pa. (See accompanying program)

Wednesday, June 24, 1936

10:00 a. m. Topic: Thermal Problems, Domestic-Commercial Field

Review of Practice in Domestic and Commercial Ice Refrigeration, by C. F. Holske

Dehumidification of Air with Coils, by William Goodman

Shell Type Porcelain Enamel Refrigerator, by F. L. Meacham and E. F. Schweller

Infiltration Characteristics of Entrance Doors, by A. M. Simpson.

District Heating Discussed at Manitoba Chapter Meeting

April 23, 1936. The fifth meeting of the Manitoba Chapter was held at the Fort Garry Hotel, Winnipeg, with 16 members and guests present.

The guest speakers were C. A. Clendenning and G. C. Davis, of the Winnipeg Heating Co., who gave an illustrated lecture on District Heating. This proved very interesting to the members, covering an unique field inasmuch as the subject discussed applied to residence and domestic hot water heating with central district steam covering a fairly large area in the City of Winnipeg.

D. F. Michie moved that a vote of thanks be given to the speakers, and Secy. C. H. Turland reports that it was decided to have the annual May meeting a social function.

Study of Combustion Results with Various Thermostats Presented to Oklahoma City Chapter

May 11, 1936. On invitation of Prof. W. H. Carson and Prof. E. F. Dawson, of the School of Mechanical Engineering, the May meeting of the Oklahoma City Chapter was held in the Engineering Auditorium of the University of Oklahoma at Norman. The members who arrived early were taken on a tour of the engineering laboratories and then had dinner at the Student Union.

Pres. F. X. Loeffler called the meeting to order at 8:10 p. m. with 35 present and the minutes of the preceding meeting were read and approved.

The members of the Committee on Standards were urged to cooperate by attending meetings.

President Loeffler announced that the next regular Chapter meeting would be held in September, and members were asked to contribute their suggestions for THE A.S.H.V.E. GUIDE for discussion at that time.

B. E. Shaw, research chief, Penn Electric Switch Co., Des Moines, Iowa, was introduced by President Loeffler and spoke on Comparative Study of Combustion Results with Various Thermostats. Great interest in this paper was shown by the discussion which followed.

Secy. E. W. Gray reports that before adjournment, the meeting was advised of the Council's recent action in regard to annual dues of Student Members and attention was called to the Society's Semi-Annual Meeting at Buck Hill Falls, Pa., June 22-24.

Ontario Chapter Holds Annual Meeting and Elects Officers

May 4, 1936. The regular annual meeting of Ontario Chapter was held at the Royal York Hotel at 6:30 p. m. with 36 members present. Following dinner, Pres. M. W. Shears stated that all limited memberships were withdrawn by request of the Council.

Announcement was made of the Semi-Annual Meeting of the Society at Buck Hill Falls, Pa., June 22-24.

The Membership Committee reported that there were 17 new members who affiliated during the past year and the Attendance Committee stated that the average attendance during the year was 47. The annual financial report of the Chapter was read by the treasurer.

W. R. Blackhall announced that J. H. Govan had received the University of Toronto's award.

The following members were unanimously elected to office:

President—Thomas McDonald.
Vice-President—G. A. Playfair.
Secretary-Treasurer—H. R. Roth.
Board of Governors—H. B. Jenney, H. D. Henion and J. W. O'Neill.

Members present discussed various sections of THE GUIDE and suggested that they be revised and that more material be contributed on engineering practice in Canada.

Mr. McDonald, seconded by O. L. Maddux, moved that the University of Toronto award be continued for two more years and this was carried.

The installation of the new officers took place and Mr. Jenney moved a vote of thanks to the outgoing officers for their splendid work during the past year.

It was the wish of the meeting that the Chapter have a golf game in the early part of June.

Life Members Honored and Control Equipment Discussed by Massachusetts Chapter

April 6, 1936. A dinner and meeting of the Massachusetts Chapter was held at the Engineer's Club, Boston, with 44 members and guests present.

Pres. W. A. McPherson read a letter from A. E. Kimberley, Washington, D. C., expressing his pleasure in presenting the discussion on air conditioning in the Department of Archives Building at the February Chapter meeting.

R. S. Franklin was appointed chairman of a committee to prepare suggestions for THE A.S.H.V.E. GUIDE.

A discussion of the Council's action in regard to limited chapter membership resulted in a decision to submit the question to the members by letter ballot to obtain an expression of opinion of the entire Chapter membership.

D. S. Boyden introduced Dr. Alice G. Bryant, who has been a Member of the A.S.H.V.E. since 1921 and who is known as an otologist and laryngologist and for her study of the related effect of atmospheric conditions. Dr. Bryant was made a Life Member of the Society at the meeting of the Council in Chicago and Mr. Boyden presented her with a certificate attesting to this fact.

W. T. Jones then introduced Alfred S. Kellogg, who has long been connected with the activities of the A.S.H.V.E. and has given generously of his time and knowledge in furthering its aims. Presentation of a Life Membership certificate was then made to Mr. Kellogg.

The speaker of the evening was J. H. Barrett, Minneapolis-Honeywell Regulator Co., who was introduced by President McPherson. Mr. Barrett discussed an instrument which controls indoor temperature from outdoor conditions, giving consideration to temperature, solar effect, wind velocity and wind direction. The instrument was explained in detail with the aid of charts, showing construction, installation, method of operation and pictures of buildings in which the equipment is installed, and later this instrument was shown in operation.

Mr. Barrett then presented a sound film on the Application of Control Equipment for Air Conditioning. Various component parts of a control system were explained in detail followed by a discussion of their coordination as required for different types of air conditioning systems. According to the report of Secy. G. B. Gerrish the explanation was comprehensive and presented valuable information.

Pacific Northwest Chapter Enjoys President's Night

April 14, 1936. The annual President's Night was held at the Washington Athletic Club, Seattle, with dinner at 6:30 and the session called to order at 7:45 p. m. The meeting was held jointly with local members of the *American Society of Mechanical Engineers* and 65 members and guests attended.

Several newcomers were greeted, including a group of five from Vancouver, B. C., who were the guests of the Pacific Northwest Chapter. The members of the A.S.M.E. were welcomed, with a response from Prof. E. O. Eastwood.

Prof. G. L. Larson, Madison, Wis., president of the A.S.H.V.E., was the guest of honor and speaker. President Larson's address on The Economics of Good Construction as Related to Residence

Heating was very ably presented and contained valuable information. The relationship of insulation and weatherstripping to the installation and operating costs of a heating plant was shown.

After an informal discussion, the meeting adjourned at 9:00 p. m., according to the report of Secy. D. C. Griffin.

Subject of Sound Interests Wisconsin Members

April 20, 1936. Thirty members and guests of Wisconsin Chapter enjoyed dinner at the City Club, Milwaukee, and others joined them in time to hear R. F. Norris, acoustical engineer in charge of research, Burgess Laboratories, Inc., Madison, Wis., who was the principal speaker of the evening.

Pres. C. H. Randolph appointed W. H. Wilson, N. B. Elliott and O. A. Trostel as tellers and, upon counting the ballots, it was found that a unanimous vote had been cast for the adoption of the proposed Wisconsin Chapter Constitution.

A letter from headquarter's office relative to limited chapter membership was read and an invitation to attend the air conditioning conference at the University of Illinois was brought to the attention of those present.

Chairman J. H. Volk presented the report of the Nominating Committee for Chapter officers for the coming year as follows:

President—M. E. Erickson.
Vice-President—W. H. Wilson.
Secretary—R. G. Koch.
Treasurer—L. A. Burch.
Board of Governors—G. L. Larson, Ernest Szekely and C. H. Randolph.

All committee chairmen were requested to submit their reports at the May meeting of the Chapter.

Mr. Norris' address on Sound provided the Wisconsin Chapter with one of the most interesting, instructive and enjoyable evenings that it has ever experienced, according to the report of Secretary Koch. Mr. Norris clearly brought out the underlying principles of a subject which is of great importance but generally least understood of the many phases of the ventilating and air conditioning industry. The audience, composed in general of those who had little or no knowledge of the subject, but with a sprinkling of those who had given the subject study or had considerable practical experience, gave the speaker unanimous and spontaneous applause.

An hour after the meeting was officially adjourned, Mr. Norris was rescued from a sizable group who from all appearances would have kept him answering questions during the remainder of the night. The Wisconsin Chapter recommends to others who desire authoritative data on the subject of Sound plus ability to convey a message in a clear, interesting and amusing manner that they obtain Mr. Norris as speaker at a chapter meeting.

Pres. G. L. Larson and W. A. Danielson Guests of Pittsburgh Chapter

May 11, 1936. The meeting was called to order at 8:05 p. m. in the University Room of Hotel Webster Hall by Pres. R. J. J. Tennant and Secy. T. F. Rockwell read the minutes of the April meeting, which were approved.

Treas. J. F. Collins, Jr., reported that the Board of Governors had authorized the transfer of $700 of the Chapter's meeting funds to a savings account in the Peoples-Pittsburgh Trust Co.

A summary of the new members who had affiliated since January was given by Chairman F. C. McIntosh of the Membership Committee.

The Air Conditioning Code Committee, under the chairmanship of George McEllroy, reported further progress and that a written statement of the committee's findings would be prepared and mailed to the members of the Chapter during the summer.

Prof. G. L. Larson, Madison, Wis., president of the A. S. H. V. E., was the guest of honor and was introduced by President Tennant. Professor Larson first mentioned briefly the recent improvement in the status of the Society and he then gave a technical discussion of the Economics of Good Construction as Related to Residence Heating.

As a concrete example, Professor Larson referred to an 8-room house having an internal cubical content of about 15,000 cu ft. He then showed how changes in construction, such as applying different types of insulation, weatherstripping and storm sash, would reduce the heat loss from the house. The analysis was then extended to show the cost of the improved construction, the return on the investment resulting from reduced fuel consumption, and possible savings in the initial cost of the heating plant. Professor Larson concluded his remarks by an analysis of the economics of humidification.

The chairman of the A. S. H. V. E. Committee on Research, W. A. Danielson, Washington, D. C., was introduced by President Tennant. Colonel Danielson first reviewed the history of research activities of the Society, including the objectives of the Research Laboratory, and outlined the present plan for organizing the research work together with plans for its financing.

The meeting was closed by W. W. Stevenson, Allegheny County Steam Heating Co., who told of the company's experiences during the St. Patrick's Day flood. Mr. Stevenson related that it was possible to keep water out of the tunnel until it was several feet deep on Penn Ave. At that time the steam was shut off and the sudden rise of cold water in the tunnel caused a rapid cooling of the lower half of the lines, while the upper half of the 24-in. lines were still near steam temperature. This severe temperature gradient, together with the buoyant force of the rising water, caused buckling of the lines which was followed by considerable distortion of the guides for the expansion joints.

Secretary Rockwell reports that the 40 members and guests thoroughly enjoyed the dinner and meeting which adjourned at 10:15 p. m.

Illinois Chapter Elects Officers and Discusses Air Conditioning

May 11, 1936. Pres. J. H. Milliken called the meeting of the Illinois Chapter to order in the Hotel Sherman at 7:40 p. m. with an attendance of 73. Since this was the annual meeting, President Milliken appointed H. P. Reid, C. W. Johnson and M. W. Bishop as tellers to count and tabulate the votes for Chapter officers.

Special communications were read including one from the Philadelphia Chapter extending an invitation to members to attend the Semi-Annual Meeting of the A. S. H. V. E. at Buck Hill Falls, Pa., June 22-24. Mention was also made of the dinner dance of the *A. S. M. E.* on May 15.

The resolution adopted by the Council of the Society in regard to the dues of Student Members was brought to the attention of the meeting.

President Milliken requested J. M. Stannard to make the announcement of the election of S. G. Neiler as a Life Member of the A. S. H. V. E. Mr. Stannard stated that Mr. Neiler, consulting engineer, Chicago, had joined the Society in 1898. Because of illness, Mr. Neiler was unable to be present, but Mr. Stannard had previously presented the certificate to him and was commissioned to report that Mr. Neiler sent greetings to the members and that he was elated at receiving this honor.

President Milliken spoke of the wonderful cooperation he had received from all of the Chapter members during the year and said that he had nothing but pleasant memories to take with him out of office.

E. W. Rietz gave the financial report for the Society's 42nd Annual Meeting, showing a balance on hand of $235.27.

Secy. L. S. Ries read a report of the Board of Governors which was accepted, and J. R. Vernon, treasurer, gave a report of the Finance Committee.

Mr. Rietz stated that the treasurer's books had been audited and found satisfactory.

Other reports were given by S. I. Rottmayer, chairman of the Entertainment Committe, E. M. Mittendorff of the Membership

Committee, and J. J. Aeberly of the Better Relations Committee.

W. L. Krogman, engineer, University of Chicago, spoke on Conditioning Operating Rooms in the Billings Hospital. He indicated that three operating rooms and one large amphitheatre in the hospital were provided with complete air conditioning, utilizing much of the existing plant at the institution. He described in detail a circular device projecting to within 10 ft of the floor from the center of the ceiling, whereby air is admitted to the room at a velocity of 350 fpm and leaves through the center of this device so that there are no drafts on the patient or the operating staff. The inadvisability of recirculating any of the conditioned air while operations were in progress was clearly set forth. After the formal talk, much interest was manifested and Mr. Krogman replied to many questions of members.

R. W. Shields, assistant engineer, office of Samuel R. Lewis, discussed Air Conditioning Requirements in a Candy Factory. He stated that the making of an assortment of candies requires a variety of air conditions and he gave the temperatures and humidities for cooling different candies, as well as for packing and storing them. Many questions were answered by Mr. Shields after the discussion and indicated interest in the able handling of the subject.

A report of the tellers stated that 81 ballots had been cast and the following officers were elected to serve Illinois Chapter for the year 1936-37:

President—J. J. Hayes.
Vice-President—L. S. Ries.
Secretary—C. E. Price.
Treasurer—J. R. Vernon.
Board of Governors—O. W. Armspach, C. E. Crone, Jr., S. I. Rottmayer.

Mr. Milliken turned the meeting over to the new president, J. J. Hayes, who responded with a spirited speech of acceptance and introduced the other officers to the Chapter, before adjournment.

Philadelphia Chapter Enjoys Visit of President Larson and Golf Meet

May 14, 1936. The Philadelphia Chapter held its annual social meeting jointly with members of the *A. S. R. E.* at the Spring Haven Country Club, and the reading of the minutes and the treasurer's report were dispensed with.

The day's activities started early and some members played 36 holes of golf. Except for a brisk movement of air, the weather was ideal. Approximately 40 members and guests teed off, and after the usual 19th hole and showers, the dinner and meeting took place.

The Philadelphia Chapter was honored in having as its guest, Prof. G. L. Larson, Madison, Wis., president of the A. S. H. V. E. Professor Larson was introduced by R. C. Bolsinger, member of the Council. In his talk, Professor Larson told in an interesting manner of the value of the Society to its members. He said he felt very proud and highly honored to have been chosen to head the A. S. H. V. E. and he concluded his talk by recounting some amusing experiences in his early contact with the Society's work.

Chairman L. A. Tucker of the *A. S. R. E.* spoke briefly and said he was delighted to be able to attend such a successful meeting and outing.

Secy. H. H. Erickson outlined plans for the Semi-Annual Meeting of the A. S. H. V. E. at Buck Hill Falls, Pa., June 22-24, and emphasized the importance of having a large Philadelphia attendance.

W. P. Culbert, golf chairman, reported the golf performances and made the awards.

Secretary Erickson reports that the day seemed to have ended too soon in adjournment by Pres. W. F. Smith at 9:00 p. m.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 46 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 15, 1936, these candidates will be ballotted upon by the council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALLEN, A. W., Sales Engr., Pease Fdry. Co., Ltd., Toronto, Ont., Can.	G. T. Wilson D. I. Paul	W. B. Pennock G. E. Cole
ALLONIER, H. R., Dist. Sales Mgr., Buckeye Blower Co., Columbus, Ohio.	K. W. Schick O. W. Motz	W. R. Stevens G. B. Houliston
BALLANTYNE, G. L., Mgr. Htg. Sales Dept., Crane, Ltd., Montreal, Que., Can.	G. L. Wiggs E. H. Gurney	A. B. Darling T. E. McGrail
BENOIST, R. E., Mgr., Benoist Bros. Supply Co., Mt. Vernon, Ill.	L. L. Benoist R. M. Rosebrough	G. W. F. Myers E. A. White
BORKAT, PHILIP, Test Engr., Air Controls, Inc., Cleveland, Ohio.	Philip Cohen G. L. Tuve	J. A. Schurman L. T. Avery
CARTER, J. H., Special Repr., Frick Co., St. Louis, Mo.	J. W. Cooper E. E. Carlson	S. L. Rolland A. A. Hoppe

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
DAVIDSON, J. C., Gas Htg. Inspector, City of Minneapolis, Minneapolis, Minn.	F. B. Rowley A. B. Algren	C. E. Lund R. E. Backstrom
DAY, I. M., Mfrs. Repr., Washington, D. C.	T. H. Urdahl C. A. Bennett (*Non-Member*)	S. J. Fentiman (*Non-Member*) J. H. Platt (*Non-Member*)
DICKENSON, F. R., Dist. Mgr., American Blower Corp., Cleveland, Ohio. (*Advancement*).	W. D. Graham G. L. Tuve	L. T. Avery H. E. Wetzell
DUFAULT, F. H., Htg. Expert, General Steel Wares, Ltd., Montreal, Que., Can.	G. L. Wiggs T. E. McGrail	C. W. Johnson A. B. Darling
ERICKSON, E. V., Engrg. Dept., Carrier Engrg. Corp., Newark, N. J.	L. L. Lewis J. H. Holton	V. S. Day Margaret Ingels
FOX, E. L., Student, Purdue University, West Lafayette, Ind.	J. D. Hoffman A. A. Potter (*A.S.M.E.*)	G. A. Young (*A.S.M.E.*) W. T. Miller (*Non-Member*)
FRANCE, C. N., Mgr., Colonial Fuel Oil, Inc., Washington, D. C.	R. H. Feltwell W. E. Kingswell	F. M. Fadeley (*Non-Member*) C. A. Blinston (*Non-Member*)
FRAZIER, J. E., Secy.-Treas., Simplex Engrg. Co., Washington, Pa.	Arthur McGonagle K. F. Treschow (*Non-Member*)	T. C. Carhart (*Non-Member*) R. C. Purdy (*Non-Member*)
FRIEDMAN, ARTHUR, Pres., Air Controls, Inc., Cleveland, Ohio.	F. R. Dickenson G. L. Tuve	H. M. Nobis W. R. Beach
GOODRAM, W. E., Partner, Goodram Bros., Hamilton, Ont., Can.	H. S. Moore W. C. Kelly	Thomas McDonald D. I. Paul
GRIEST, KERMIT, Sheet Metal Worker, Frank Limbach Co., Pittsburgh, Pa.	W. H. Osterle C. M. Humphreys	John Proie T. F. Rockwell
HALE, F. J., Mgr. & Partner, Empire Sheet Metal Works, Ltd., Vancouver, B. C.	R. E. Johnston Walter Leek	J. P. Mackenzie (*M.E.I.C.*) S. A. Lake (*R.P.E.*)
HARTLINE, W. R., Vice Pres. & Treas., Combustioneer Stoker Corp., Washington, D. C.	A. E. Beitzell, Jr. T. H. Urdahl	W. E. Kingswell G. S. Frankel
HUGHES, W. U., Managing Director, The Lewis-Brown Co., Ltd., Montreal, Que., Can.	G. L. Wiggs Leo Garneau	E. H. Gurney C. W. Johnson
HUNT, MACDONALD, Mfrs. Agent, Baltimore, Md.	W. A. Danielson R. H. Feltwell	T. H. Smoot T. H. Urdahl
JOSEPHSON, SIMON, Plbg. & Htg. Engr., Astor Plbg. & Htg. Corp., Brooklyn, N. Y.	M. C. Giannini W. R. Zuhlke	A. A. Adler J. H. Seckler (*Non-Member*)
KIMMELL, P. M., Engr., L. R. Krumm Co., Columbus, Ohio.	A. R. MacMillan J. D. Hoffman	C. E. Lewis W. T. Miller (*Non-Member*)
KUBASTA, R. W., Sales Engr., Bryant Air Cond. Corp., New York, N. Y.	W. E. Stark D. W. Nelson	G. L. Larson Ernest Szekely
LAMONTAGNE, A. F., Sales Mgr., The Gurney Fdry. Co., Ltd., Montreal, Que., Can.	G. L. Wiggs E. H. Gurney	A. W. Givin Leo Garneau
LANGE, R. T., Engr., Hartzell Propeller Fan Co., Piqua, Ohio.	W. A. Rowe A. G. Sutcliffe	J. M. Frank Frederick Charaway (*S.A.E.*)
LEDGETT, F. D., Student, University of Toronto, Toronto, Ont., Can.	D. O. Price Thomas McDonald	M. W. Shears J. H. Fox
MARCONETT, V. G., Engr. & Factory Supt., The Farquhar Furnace Co., Wilmington, Ohio	I. B. Doyle (*Non-Member*) W. R. Pyle (*Non-Member*)	E. J. Rose (*Non-Member*) R. J. Miars (*Non-Member*)
MARTIN, LEONARD, Sales Engr., H. L. Peiler & Co., Montreal, Que., Can.	A. S. Leitch C. W. Johnson	W. R. Blackhall G. L. Wiggs
MCMAHON, J. E., Student, University of Illinois, Urbana, Ill.	W. H. Severns M. K. Fahnestock	P. E. Mohn (*A.S.M.E.*) O. A. Leutwiler (*A.S.M.E.*)
MOULD, D. E., Mgr., J. W. Mould & Son, Plbg. & Htg., Edmonton, Alta., Can.	C. H. Turland A. W. Givin	E. H. Gurney G. L. Wiggs
PARK, H. E., Sales, Shaw-Perkins Mfg. Co., Pittsburgh, Pa.	W. H. Osterle C. M. Humphreys	John Proie T. F. Rockwell
POLING, D. B., Engr., Lamneck Products, Inc., Columbus, Ohio.	W. E. Stark W. H. Lauer (*Non-Member*)	G. C. Mason (*Non-Member*) E. S. Buck (*A.S.M.E.*)
RODEE, E. J., Engr., John B. Pierce Lab. of Hygiene, New Haven, Conn.	C.-E. A. Winslow C. W. Brabbee	L. E. Seeley F. V. Hartman (*A.S.M.E.*)
SAPP, C. L., Sales Mgr., Farquhar Furnace Co., Wilmington, Ohio.	I. B. Doyle (*Non-Member*) W. R. Pyle (*Non-Member*)	E. J. Rose (*Non-Member*) R. J. Miars (*Non-Member*)
SCOTT, L. G., Asst. to Supt. of Bldgs., Hudson's Bay Co., Winnipeg, Man., Can.	C. H. Turland J. B. Steele	R. L. Kent A. H. Doe (*Non-Member*)
SHILLING, H. C., Sales Engr., Barber-Colman Co., Chicago, Ill.	C. J. Braatz D. J. Stewart	R. P. Dewey J. E. McClellan
SOUTHMAYD, R. T., Sales Engr., American Blower Corp., Cleveland, Ohio.	F. R. Dickenson W. R. Rhoton	L. T. Avery H. E. Wetzell
TOLHURST, G. C., Htg. Expert, The Gurney Fdry. Co., Ltd., Montreal, Que., Can.	G. L. Wiggs E. H. Gurney	A. W. Givin Leo Garneau
TURNER, H. S., JR., Student, Agric. & Mech. College of Texas, College Station, Texas.	W. E. Long A. V. Brewer (*Non-Member*)	V. M. Faires (*A.S.M.E.*) C. W. Crawford (*A.S.M.E.*)
VOYE, V. J., Sales, Richardson & Boynton Co., Boston, Mass.	L. A. Brissette W. F. Gilling, Jr.	J. J. Murray R. S. Franklin
WARD, J. J., Pres. & Mgr., Wenzler & Ward, Inc., Seattle, Wash. (*Reinstatement*)	C. F. Twist D. C. Griffin	W. E. Beggs C. W. May
WHERMAN, W. J., Student, University of Minnesota, Minneapolis, Minn.	F. B. Rowley A. B. Algren	J. V. Martenis C. F. Shoop (*Non-Member*)

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
WIGGS, G. L., Consulting Engr., Montreal, Que., Can. (*Advancement*).	E. H. Gurney A. W. Givin	W. W. Timmis A. V. Hutchinson
WOOLLEY, J. H., Vice Pres., Woolley Coal Co., Inc., Maplewood, N. J.	Pensyl Mawby G. W. Emery	E. W. Peters W. C. Lawson
YATES, J. E. JR., Service & Installation Mgr., Automatic Htg. Co., Ltd., Winnipeg, Man., Can.	William Glass J. E. Yates	D. G. Michie H. R. Eade

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and ballotted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ALLEN, D. M., Dist. Mgr., Ilg Electric Vtg. Co., Kansas City, Mo. (*Reinstatement & Advancement*).

BARNARD, M. E., Sales Engr., Carrier Engrg. Corp., Philadelphia, Pa. (*Reinstatement*).

BROCHA, J. F., Buyer, Plbg. Htg. Dept., Montgomery Ward Co., Chicago, Ill.

CARRIER, E. G., Sales Engr., Carrier South Africa (pty) Ltd., Johannesburg, Transvaal, South Africa. (*Advancement*).

CROSBY, E. L., Consulting Engr., Henry Adams, Inc., Baltimore, Md.

CUMMING, F. J., Pres., Beecher-Cumming, Inc., Minneapolis, Minn.

DOWNS, C. R., Vice Pres., Weiss & Downs, Inc., New York, N. Y.

ELWOOD, W. H., Br. Mgr., Holland Furnace Co., Ithaca, N. Y.

FOOTE, E. E., Chief Engr. & Gen. Supt., Consumers Central Htg. Co., Tacoma, Wash.

FORSLUND, O. A., Partner & Mgr., Forslund Pump & Machinery Co., Kansas City, Mo.

FRENTZEL, H. C., Chief Engr., The Heil Co., Milwaukee, Wis.

GILL, E. F., Chief Draftsman, Drayton Regulator & Instrument Corp., West Drayton, Middlesex, England.

GREBEN, DAVID, Consulting Engr., New York, N. Y.

HOCKENSMITH, F. E., Chief Htg. Engr., Lennox Furnace Co., Syracuse, N. Y.

HOWE, W. W., Div. Sales Engr., Pacific Gas & Elec. Co., Sausalito, Calif. (*Reinstatement & Advancement*).

KNAB, E. A., Htg. Contractor, E. A. Knab Htg. Contractors, Milwaukee, Wis. (*Reinstatement*).

LAUFKETTER, F. C., Chief Engr. & Supt., Jefferson Hotel, St. Louis, Mo.

LUND, C. E., Warm Air Htg. Inspector, City of Minneapolis, Minneapolis, Minn. (*Advancement*).

MCKITRICK, W. D., Htg. Engr., Mills, Rhines, Bellman & Nordhoff, Inc., Toledo, Ohio.

MILENER, E. D., Secy., Industrial Gas Section, American Gas Assn., New York, N. Y.

MILLER, G. F., Mfr's. Agent, C. A. Dunham Co., Washington, D. C.

NEST, R. E., Asst. Chief Engr., Anchor Post Fence Co., Baltimore, Md.

NORDINE, L. F., Br. Mgr., The Trane Co., Washington, D. C. (*Reinstatement*).

ORR, G. M., Consulting Engr., G. M. Orr & Co., Minneapolis, Minn.

PATERSON, F. C., JR., Pres. & Treas., F. C. Paterson & Co., Inc., Bradford, Pa. (*Reinstatement & Advancement*).

PAUL, D. I., Sales Engr., Gurney Fdry. Co., Ltd., Toronto, Ont., Can. (*Advancement*).

PIETSCH, J. A., Consulting Engr., J. A. Pietsch, Inc., New Brighton, S. I., N. Y.

PRUDEN, BRADLEE, Barber-Colman Co., Cleveland, Ohio.

RHOTON, W. R., Pres., The W. R. Rhoton Co., Cleveland, Ohio.

SCARLETT, W. J., Sales Engr., Carrier Engrg. Corp., Kansas City, Mo.

SCHURMAN, J. A., Air Cond. Engr., York Ice Machinery Corp., Cleveland, Ohio. (*Advancement*).

SMITH, F. J., Engr., Northern Air Cond. Corp., Newark, N. J. (*Reinstatement*).

SMITH, R. J., Mgr., Smith & Elston, Timmins, Ont., Can.

SWISHER, S. G., JR., Sales Engr., The Trane Co., La Crosse, Wis. (*Advancement*).

WEISS, C. A., Mgr., Kornbrodt Kornice Ko., Kansas City, Mo. (*Advancement*).

WHITTEN, H. E., Pres. & Treas., H. E. Whitten Co., Boston, Mass. (*Reinstatement*).

ASSOCIATES

BALL, WILLIAM, Pres., Interstate Htg. & Plbg. Co., Kansas City, Mo.

BERGLUND, N. W., Draftsman, York Ice Machinery Corp., Los Angeles, Calif.

BUCK, D. T., Pres., Buck Engrg. Co., Inc., Freehold, N. J.

CLEVELAND, C. C., Member of Firm, Johnson & Cleveland, Bradford, Pa.

DANIELSON, E. B., Owner & Mgr., Air Cond. Co., Russell, Kans.

EARL, WARREN, Sales Engr., W. C. Dance, Inc., Oklahoma City, Okla.

FOSS, E. R., Southern Mgr., The Powers Regulator Co., Atlanta, Ga.

GARNETT, R. E., Sales Engr., Standard Asbestos Mfg. & Insulating Co., Kansas City, Mo.

HOBBS, W. S., Owner & Mgr., Wm. S. Hobbs, Swarthmore, Pa.

HOLMES, P. B., Br. Mgr., National Radiator Corp., Washington, D. C.

ISETT, W. M., Pres., C. B. Isett & Son, Inc., Chicago, Ill.

TUCKER, T. T., Chief Engr., Armor Insulating Co., Atlanta, Ga.

VAUGHN, F. R., Vice Pres., Green Fdry. & Furnace Wks., Des Moines, Iowa.

JUNIORS

DRAKE, G. M., Vice Pres., Geo. H. Drake, Inc., Buffalo, N. Y.

DULLE, W. L., Asst. Secy., E. E. Souther Iron Co., St. Louis, Mo.

FITCH, H. M., Grad. Student, University of Ky., Lexington, Ky.

GELTZ, R. W., Air Cond. Engr., York Ice Machinery Corp., Cleveland, Ohio.

GRABER, ERNEST, Engr., Minneapolis-Honeywell Reg. Co., New York, N. Y.

KARAKASH, T. J., Equipment Engr., Ford Motor Co., Istanbul, Turkey.

LARSON, C. P., Sales Engr., The Insulite Co., Chicago, Ill.

LOCKHART, W. R., Sales Engr., York Ice Machinery Corp., Washington, D. C.

MACROW, LAWRENCE, Engr., Carrier Engrg. Corp., Newark, N. J.

MIDDLETON, D. K., Sales Engr., Johnson Service Co., Oklahoma City, Okla.

SCHERRER, L. B., Engr. & Estimator, Housing Service, Inc., St. Louis, Mo.

WENDT, E. H., Draftsman & Estimator, Wendt & Crone Co., Chicago, Ill.

STUDENTS

FAY, D. P., Student, Pratt Inst., Brooklyn, N. Y.

JACOBI, B. A., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.

JOHNSON, W. G., Student, University of Illinois, Urbana, Ill.

SCHERMER, RICHARD, Student, Armour Inst. of Tech., Chicago, Ill.

VAN NUYS, J. C., Student, Pratt Inst., Brooklyn, N. Y.

THE COST OF SERVICE!

IN YEARS GONE BY

80% of all service calls were directly or indirectly caused by mechanical failure of the expansion valve or its inability to control under varying conditions. Excessive service expense was directly responsible for the failure of most refrigeration companies.

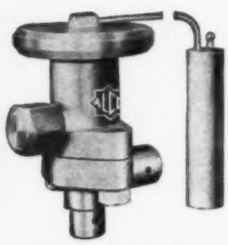
NOW

ALCO

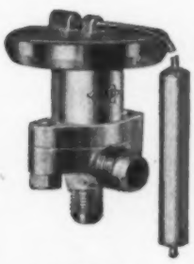
Series "T"

THERMO VALVES

have changed the Service Situation



TYPE TCL



TYPE TL

They are built to "stand the gaff" and are guaranteed to do it.

Progressive refrigeration companies are using Alco Series "T" Thermo Valves and making money.

Alco Series "T" Thermo Valves for Sulphur Dioxide, Methyl Chloride and Freon are startling innovations. No other valves compare in simplicity of design, rugged construction, ease of installation, freedom from trouble, accuracy of control, or choice of sizes to accommodate all capacities required in modern refrigerating systems.

THEY COST FAR LESS IN THE FINAL ANALYSIS

For further details, ask for Bulletin 144C.

ALCO VALVE CO., INC.

2626 Big Bend Blvd., St. Louis, Mo., U.S.A.

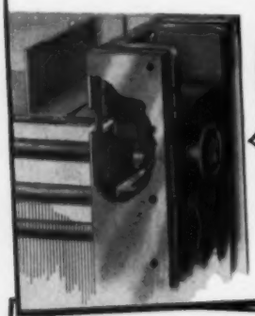
New York Chicago Los Angeles San Francisco

—Automatic Refrigeration Control Valves for All Refrigerants and Applications—

Mr. Engineer!

GET THIS EXCLUSIVE STORY on TRANE TYPE "E" BLAST COILS

EXCLUSIVE TRANE GUIDE FLANGE



This cut-away view of the Coil shows the patented guide flange construction. It permits the Coil core to expand and contract independently of the casing, relieving all stress and strain. A sure, positive method of preventing leaks. There is nothing like it on the market.

EXCLUSIVE TRANE FIN CONSTRUCTION

Due to the patented process of mechanically bonding the tubes to the fins without welding, soldering or brazing, TRANE Coils are more efficient, offer a greater free area, and are lighter in weight. These are distinct advantages found only in TRANE Blast Coils.

AN EXCLUSIVE HELP IN SELECTION

TRANE Blast Coils are manufactured in a wide range of sizes and capacities for both high and low pressure steam. This, plus the fact that TRANE Bulletins are authoritative and easy to understand makes the Line exceedingly popular. It is easy to select exactly the right TRANE Coil for every requirement.

Ask Trane first

--ABOUT ALL INDUSTRIAL AND COMMERCIAL AIR CONDITIONING PROBLEMS INVOLVING THE USE OF COOLING COILS, COOLING UNITS, PRODUCT COOLERS, OR UNIT HEATERS OR HEATING SPECIALTIES.

THE TRANE COMPANY

BRANCHES IN ALL PRINCIPAL CITIES

LA CROSSE, WISCONSIN

In Canada—Mowat & King Sts. W.—Toronto

TRANE

The Editor's Page—

[*Continued from page 31, front section*]

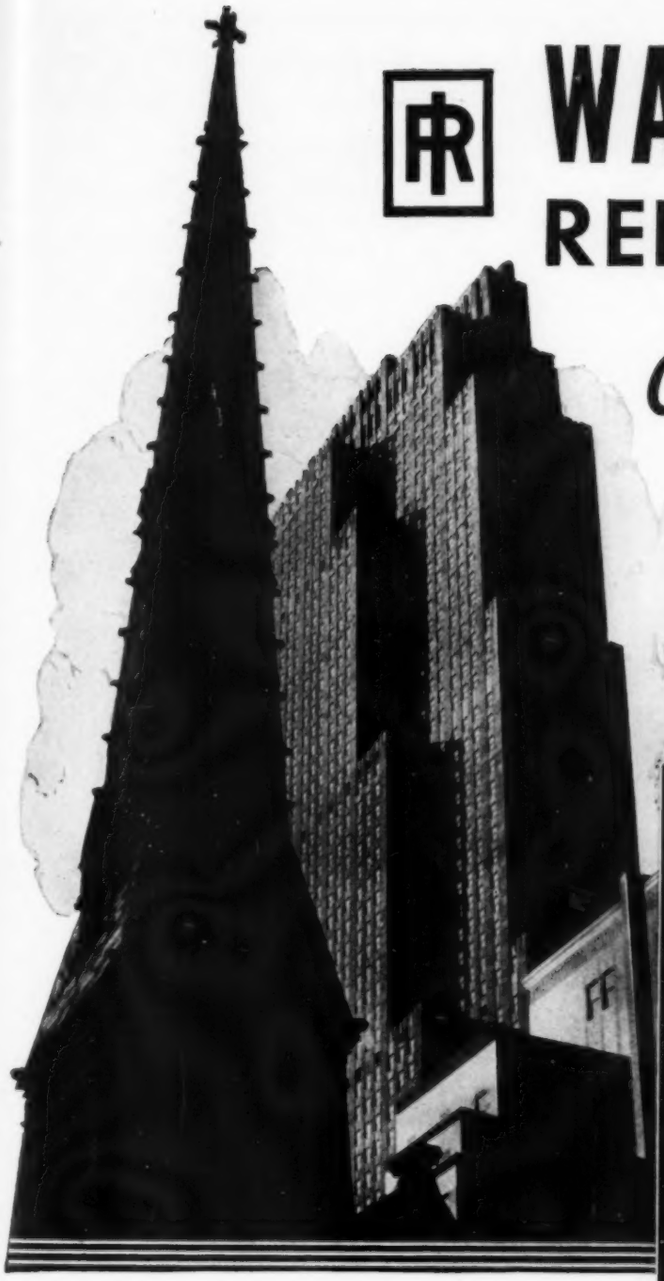
vention at Dallas last month. "No two installations are ever exactly alike, for the conditions to be overcome are never exactly alike in any two instances," he said. "An air conditioning installation results from the pooling of many single items of equipment into an assembly by a competent engineer in such manner as to produce a specified atmospheric result within an enclosed space," he stressed, warning his listeners against the use of the term "air conditioning" applied to equipment which does only a part of the complete job.

■ The company founded by Joseph Nason, pioneer of American steam heating practice, has recently been acquired by Detroit interests, and the present address of the Nason Co. is 1050 Mt. Elliott Ave., Detroit. . . . Mr. Nason, who was born in Boston in 1815, is credited with making the first globe valve, devised by him to take the place of the cock, the only stop available which could be put into a straight line of pipe. Of uncommon inventive ability, he also devised the taper thread for pipe joints, the pitch being ¾ in. to a foot, about 1852 devised and introduced malleable iron fittings as a substitute for cast iron, developed the cast iron radiator, built the first steam trap in America, and invented the balanced regulating valve.

■ Willis H. Carrier presented his classic paper on air conditioning entitled "Rational Psychrometric Formulae" at the annual meeting of the American Society of Mechanical Engineers held in New York City twenty-five years ago next December. The following psychrometric principles were demonstrated experimentally by a method described in the appendix to his paper: (1) When dry air is saturated adiabatically the temperature is reduced as the absolute humidity is increased, and the decrease of sensible heat is exactly equal to the simultaneous increase in latent heat due to evaporation. (2) As the moisture content of air is increased adiabatically, the temperature is reduced simultaneously until the vapor pressure corresponds to the temperature, when no further heat metamorphosis is possible. This ultimate temperature may be termed the temperature of adiabatic saturation. (3) When an insulated body of water is permitted to evaporate freely in the air, it assumes the temperature of adiabatic saturation of that air and is unaffected by convection; *i.e.*, the true wet bulb temperature of air is identical with its temperature of adiabatic saturation. From the above is deduced: (4) The true wet bulb temperature of the air depends entirely on the total of the sensible and the latent heat in the air and is independent of their relative proportions. In other words, the wet bulb temperature of the air is constant, providing the total heat of the air is constant.

■ *R. D. Touton of Philadelphia, Technical Director of Bayuk Cigars and a Member of H. P. & A. C.'s Board of Consulting and Contributing Editors, stressed the importance of engineering in every installation for air conditioning when he visited the editorial offices on his way east after a California business trip. Most unsatisfactory systems fall down in the matter of proper air circulation to prevent drafts or dead areas, inadequate provisions for zoning when it would be desirable for control, and impossibility of using outside air for economy in operating costs when outside conditions are favorable, he believes. Responsibility of the designer should not end when the job is installed, Mr. Touton feels, but he should assure himself that the operator is fully instructed and that the system is operated as intended.*

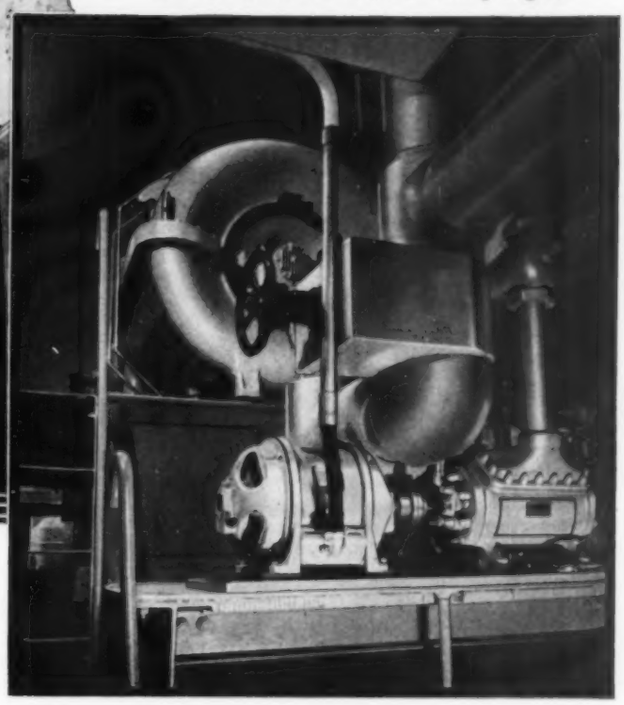
■ A special award was made to the Insurance Building, Omaha, Nebr., in the industrial modernization contest conducted by "Forbes Magazine" for an outstanding instance of what can be done by modernization to recapture tenants for an old and obsolete office building. . . . Air conditioning was the feature of the Insurance Building's modernization program.



WATER-VAPOR REFRIGERATION

On the 58th floor of the RCA Building

Francis Y. Joannes, *Architect*
Jaros, Baum & Bolles, *Consulting Engrs.*



Offices on four floors, 750 feet above the street in the R.C.A. Building, New York City, are cooled with the aid of water chilled by a 180-ton Ingersoll-Rand Centrifugal Water-Vapor Refrigerating Unit which is beginning its third season.

Using water as the only refrigerating medium, this unit represents the ultimate in safety to the thousands of people in this modern sky-scraper office building. It assures freedom from restrictive ordinances covering the use and storage of chemical refrigerants. Leakage, storage and replacement of refrigerant are forever eliminated.

In addition, I-R Units assure their owners: absence of vibration; capacity unimpaired by wear; self-regulation of power input with load; and unusual overload capacity.

WATER IS THE ONLY REFRIGERATING MEDIUM

Birmingham, Boston, Buffalo, Butte, Chicago, Cleveland, Dallas, Denver, Detroit, Duluth, El Paso, Houston, Knoxville

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develop the most complete line of Unit Heaters made

1. TESTING A TRANE UNIT HEATER COIL WITH SPRAY NOZZLES IN THE DEVELOPMENT OF A SPECIAL HUMIDIFYING UNIT.

2. A TRANE UNIT HEATER UNDER AN OFF-AND-ON BREAKDOWN TEST WITH ALTERNATE STEAM AND COLD WATER RUNNING THROUGH THE COIL.

3. TRANE UNIT HEATER COIL ON A WIND TUNNEL TEST WHERE CAPACITIES ARE BEING CHECKED.

There is a reason why the TRANE Unit Heater Line is more complete than any other made. In the TRANE laboratories, hundreds of tests annually are used to develop, check and refine the entire Line. And these tests are under the supervision of skilled, experienced Engineers. They have been actively engaged for many years in the design of Unit Heaters for every industrial and commercial application.

The results of their work may be seen not only in installations where the Units are used for straight heating, but also to solve such individual problems as concrete drying, fog elimination, frost elimination, and many other unusual applications. And 1936 will also see further development in the TRANE Line.

Announcements will be made later which will be interesting to everyone specifying Unit Heater Equipment.

Ask Trane first

The TRANE Company also manufactures a complete line of Blast Coils, Cooling Coils, and Industrial and Commercial Air Conditioning Equipment.

ABOUT THE ENGINEERING SERVICE YOU CAN OBTAIN TO ASSIST YOU IN LAYING OUT UNIT HEATER SYSTEMS. SKILLED ENGINEERS WILL GIVE YOU EVERY COOPERATION.

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TRANE

Booklets, Reports and Papers

Physical Factors Affecting Comfort

The physical factors affecting comfort and involved in air conditioning and heating were outlined by Prof. A. P. Kratz at the conference on air conditioning at the University of Illinois last month; his paper summarized much of the research on heating and cooling that has been done at the University.

Prof. Kratz pointed out that the progress of civilization has been marked to a considerable extent by man's success in his efforts to control his natural environment, and that his earliest efforts in this direction were toward the conservation of bodily heat. He then described the development of theories of comfort and the conditions required for comfort, and said that the proper function of a heating or cooling plant is not to warm or cool the body, but to produce an environment in which the body is able to lose the amount of heat corresponding to that generated, without any conscious bodily adjustments having to be made.

The factors affecting heat losses from the body were explained, the relation between type of heating system and comfort were discussed and the effect of cold walls and windows was given attention. It was mentioned that higher air temperatures are demanded to offset cold exposures and that the presence of hot walls and ceilings makes necessary the use of lower air temperatures for the same degree of comfort.

Temperature Gradients

The effect of temperature gradients in rooms was brought out and readings typical of different types of heat transmitting units were given. A great deal of research has been done at the University on temperature gradients of various types of radiators and convectors. Results of this work indicate that a thermometer placed at the 5 ft level may not give a true indication of comfort, depending upon the type of heating plant and other factors, and that it would be better practice to locate a thermometer at the 30 in. level. The same principles apply even more forcibly to the location of the thermostat, according to the speaker. A thermostat at the 5 ft level cannot act to admit more heat to the room until the air at that level has cooled to the thermostat setting. This means a long "off" period with the collection of a deep pool of cool or cold air near the floor with a very pronounced temperature gradient. With gas or oil fired systems this is more prevalent than with coal fired.

Humidification was discussed and it was stressed that medical opinion concerning the physiological effects of atmospheres of low relative humidity was not entirely unanimous, and that hence, the matter of comfort affords the best argument in favor of adequate humidification.

Importance of Building Construction

Prof. Kratz devoted attention to the importance of good building construction and its effect on heating and cooling, and it was stated that the features of good construction such as air tight walls, heat insulation, weather stripped windows and storm sash all have significant bearing. In concluding, he stated that considerable efforts have been directed toward the design of the heating and cooling plant, but until recently not enough attention has been paid to the structure housing this plant, and that the air conditioning engineer should arouse proper interest in good construction, as it is inseparable from proper operation of the heating and cooling equipment.

[Other of the papers presented at the Conference will be summarized in future issues.]

All in a single compact unit!

Cleaned Humidified Tempered Circulated **AIR**

Economical Steel Boiler **HEAT**

Year 'round **HOT WATER** SUPPLY with no storage tank

The New FITZGIBBONS BOILER-AIR CONDITIONER

With any good oil burner this new FITZGIBBONS development comprises a unit unique in the multiplicity of services it renders, while requiring the minimum of basement floor space.

It fits beautifully in any scheme of basement design or decoration, and marks any home it serves as *modern* to the final degree.

Get full information about this revolutionary residential unit. Write—NOW!

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BRANCHES AND REPRESENTATIVES IN PRINCIPAL CITIES

50 YEARS STEEL HEATING BOILERS

Wolverine Refrigeration Tubing

When you use Wolverine Refrigeration Tubing or Wolverine Deoxidized Water Tubing for refrigeration or air conditioning work, you can rely 100% on the quality of the installation because:

1. It is made of 99.9% pure copper which defies rust or corrosion.
2. It is produced by the special Wolverine extrusion process which assures a clean bright tube with a mirror-like inside finish—no scale, shop dirt or sediment to clog fuel, oil or water lines.
3. Has uniform soft temper to facilitate cutting, bending, flaring and installation.
4. Has uniform wall thickness and is accurately sized to assure tight joints with either flared or soldered fittings.
5. Is always available in whatever size you need at your local Refrigeration supply house.

Standardize on Wolverine Deoxidized Copper Water Tubing for all automatic heat and air conditioning installations.

WOLVERINE TUBE CO.

SEAMLESS COPPER BRASS & ALUMINUM

Code for Ventilation and Air Conditioning

Following is Part XIII of the heating code of Minneapolis, Minn., dated 1936, covering ventilation and air conditioning, and reproduced through the courtesy of Louis Clousing, Inspector of Buildings:—

PART XIII—VENTILATION AND AIR CONDITIONING

Section 1301. **Definitions:** For the purposes of this ordinance the following definitions shall govern as to the meaning of the several terms and expressions so defined, wherever said terms and expressions are employed in this ordinance:

1301.1 **Ventilation:** The term ventilation shall be taken to mean the process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

1301.2. **Air Conditioning:** The term air conditioning shall mean provision for the simultaneous automatic control of the temperature, motion, and humidity and a reduction in the dust content and odors for air employed in the ventilation of rooms.

Three classifications of air conditioning shall be recognized under this ordinance as follows:

(a) Winter air conditioning which shall include the proper distribution of the cleaned, humidified and heated air to and within the spaces to be conditioned.

(b) Summer air conditioning which shall include the proper distribution of the cleaned, dehumidified and cooled air to and within the spaces to be conditioned.

(c) All year air conditioning which shall provide for all of the services enumerated in (a) and (b) above.

Section 1302. **Information for Checking:** The plans and specifications of all air conditioning installations submitted for approval under this ordinance shall set forth the heat transmission coefficients for barriers and the amounts and sources of heat loss and gain. The design temperatures and relative humidity shall be recorded for air outside and inside, also for condensing air or water, and for the heat transferring medium.

These governing design factors and temperatures shall be such as to produce an effective result at least equal to those set forth in this ordinance.

Section 1303. **Design Coefficients for Heat Tranfer Through Construction Barriers and for Heat Gains from People or Appliances:** Design coefficients for heat transfer through construction barriers and for heat gains from people or appliances shall conform to the standards incorporated in Chapters 1 to 13 inclusive, Volume 13, *American Society of Heating and Ventilating Engineers Guide*, 1935 Edition.

1303.1 **Outside and Inside Design Temperatures and Relative Humidity in Winter:** Outside and inside design temperatures in winter shall be as set forth in Part II of this ordinance.[1] The design inside relative humidity shall be 50 per cent when the outside condition is 30 F and 40 per cent Rh.

[1]Part II states that the outside air temperature for design purposes shall be taken as 20 F below zero, and the inside temperature (at the breathing line) shall be taken from the following table:

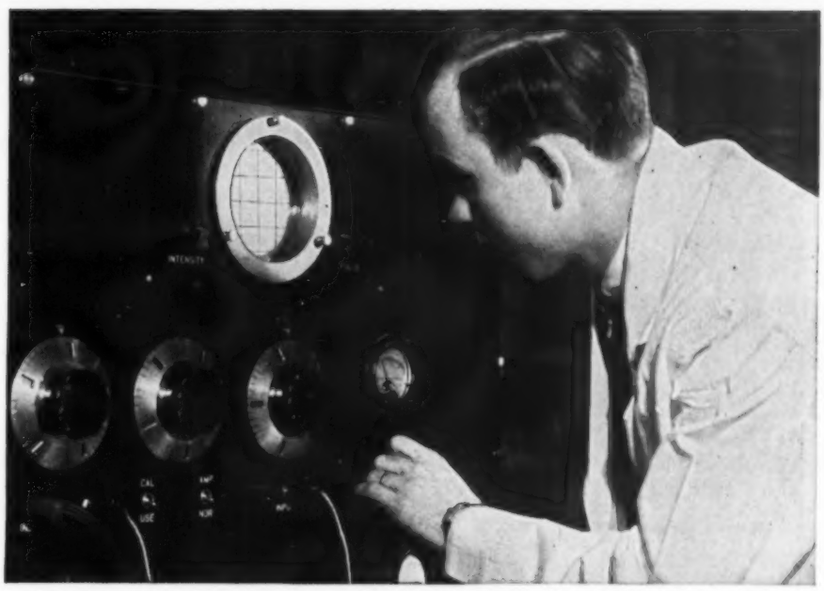
Dwellings as defined by the State Housing Act, Offices, Retail Stores, Schools, Places of Assembly, Theaters, Dance Halls, Restaurants or buildings of similar occupancy and all employment areas of more or less sedentary occupations			70 F
Industrial spaces	Employment areas	light labor	65 F
		heavy labor	55 F
	Storage areas		None
Garages	Storage		40 F
	Employment areas		60 F

Provided, however, that in industrial and storage spaces where the manufacture, processing or storage of materials or products requires a temperature lower than those above specified, special permission may be granted by the Inspector of Buildings to use such lower temperature for design purposes as conditions may warrant.

NOW SEE WHAT CHRYSLER OFFERS

IN AUTOMATIC HEATING AND AIR CONDITIONING

Airtemp's Complete Line of Economical, Modern, Dependable Equipment.. from one Reliable Source



AIRTEMP'S HIGH EFFICIENCY is the result of years of testing and development

THE ENTRANCE of Chrysler's Airtemp into this new field opens big opportunities for men who qualify as distributors and dealers of today's most complete line of modern automatic air conditioning and home heating equipment.

The Chrysler-engineered Airtemp line covers a wide range of simplified, dependable systems . . . winter, summer, all year air conditioners for any home or building . . . self-contained air conditioners for a single room or office . . . a NEW Airtemp Automatic Oil Burner . . . a complete oil-fired Boiler Burner . . . a complete Gas-Fired Boiler. Most modern in design.

Home Air Conditioning

Airtemp Air Conditioners offer a 100% selling story on comfort and health. Airtemp ventilates . . . heats, humidifies . . . cools, dehumidifies . . . filters . . . circulates air. Airtemp provides a "Split System" . . . heating and air conditioning units separate, but working in unison. Airtemp also provides a NEW Direct-Fired system . . . a winter heating and air conditioning system in one. What's more, there's a NEW Airtemp Summer Cooling System to match . . . entirely Chrysler-built . . . requiring only simple connections.

FOR THE BUSINESS FIELD the Airtemp line ranges from a ½-ton self-contained unit . . . to large tonnage central station systems.

Dependable Operation

Airtemp Air Conditioners are operating successfully . . . in residences . . . theatres, shops, hotels, dept. stores. Airtemp Automatic Heating Units, too, are operating efficiently and economically in all sizes of homes.

Partners Wanted

Men of responsibility and character are wanted as distributing partners. Airtemp engineers back you to the limit. Your sales efforts are backed by a typical Chrysler national advertising campaign. If you feel you qualify for this Chrysler opportunity . . . send the coupon at once . . . or wire.

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AUTOMATIC HOME HEATING & AIR CONDITIONING SYSTEMS

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DETROIT, MICH.

Gentlemen:—Send me details of the Chrysler Corporation Airtemp Proposition.

Name_____

Address_____

City_____ State_____

SUBSIDIARY OF CHRYSLER CORPORATION

1303.2 **Outside Design Temperature and Relative Humidity in Summer:** The design dry bulb and wet bulb temperature shall not be cooler than 95 F dry bulb and 75 F wet bulb respectively. Approximately 40 per cent relative humidity.

1303.3. **Inside Design Temperature and Relative Humidity in Summer:** Three classifications for inside conditions shall be recognized by this ordinance as follows:

Type A: In spaces which are occupied continuously for more than three hours or where exceptional cooling is desired, the assumed inside design temperature and relative humidity shall be not more than 78 F dry bulb and 67 F wet bulb. Approximately 56 per cent Rh.

Type B: In spaces of normal occupancy or not more than three hours, the assumed inside design temperature and relative humidity shall be not more than 80 F dry bulb and 68 F wet bulb. Approximately 53 per cent Rh. This classification will be standard for most installations.

Type C: In spaces were the occupancy is less than one hour or such installations where the design requirements are less exacting, the assumed inside design temperature and relative humidity shall be not more than 82 F dry bulb and 69 F wet bulb. Approximately 52 per cent Rh.

1303.4. **Infiltration and Leakage:** Design capacity to care for infiltration of air from outside shall be provided in all heating and cooling systems except that in a system so designed that a positive pressure is maintained within the room or space to be heated or cooled the heat loss or gain due to infiltration and air leakage may be omitted in design calculations.

The coefficients and data set forth in Chapter 6, Volume 13, *American Society of Heating and Ventilating Engineers Guide*, 1935 Edition, shall be used in computing infiltration losses or gains.

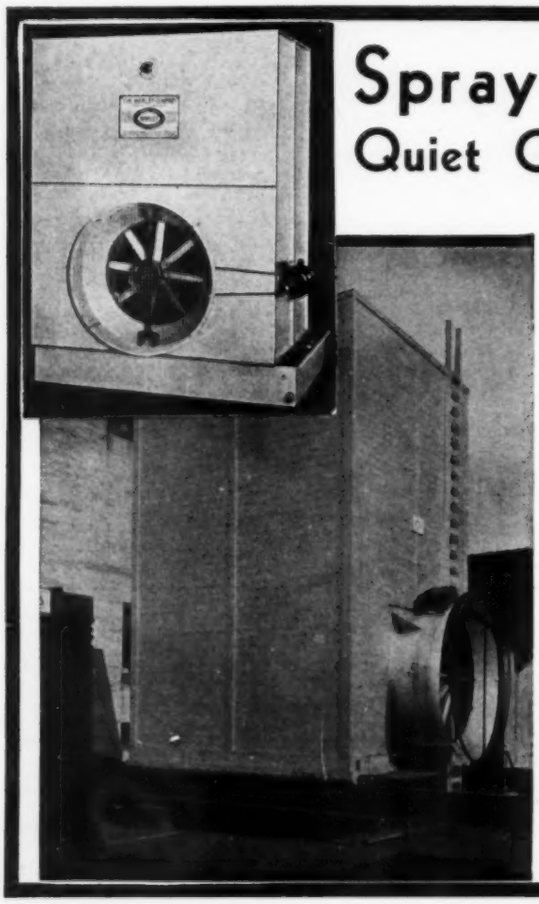
The above comprise sensible heat only. The latent heat or moisture gain, by far the more serious problem in dealing with infiltration, must be computed for the exact temperature and moisture differences.

1303.5. **Design Allowances for Heat Gain Due to Sunlight:** The heat gain from sunlight varies from hour to hour on various exposures, and in order to get a correct total heat gain for any hour of the day its value on all surfaces exposed to sunlight must be added to the sum of all other heat gains occurring at the same hour. The highest total heat gain may occur at the time of highest conductance load, or at the time of highest occupancy load, or at the time of greatest sun effect on the largest exposed wall or glass areas. Final calculations shall be based on the hour giving the maximum total of all heat gains, taking into account the time lag factor due to heat capacity of the structure and the consequent time lag in the transmission of heat. The tables, curves and data incorporated in Chapter 8, Volume 13, *American Society of Heating and Ventilating Engineers Guide*, 1935 Edition, shall be used in such calculations.

1303.6. **Design Allowance for Shading from Sunshine:** Design allowance for shading from sunlight shall be made only when the plans and specifications specifically show such allowance, and give assurance that the owner is cognizant of the reduction in capacity on this account.

1303.7. **Design Air Quantity:** (a) For winter air conditioning the design air quantity shall be not less than the requirements of the Minneapolis Building Code adopted March 29, 1934. In all cases a minimum air circulation of four (4) air changes per hour or twenty-five (25) cubic feet per minute per person, whichever is greatest, shall be provided.

(b) For summer air conditioning the design air quantity shall be sufficient to extract the surplus heat at the air inlet and outlet temperature specified and in addition shall be not less than the minimum requirements specified for winter air conditioning above.



Spray Nozzle Distribution
Quiet Operation - - Drift Elimination

These are three of the reasons why Marley towers are used from coast to coast in air conditioning work. They meet the need for more efficient water cooling —more practical design.

Write for complete details.

THE MARLEY COMPANY
1915 Walnut, Kansas City, Mo.

MARLEY
Forced Draft Cooling Towers

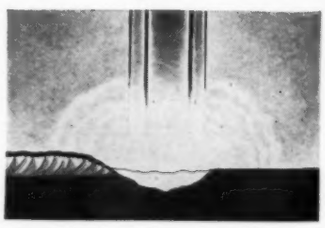
Equipped with Marley Patented Spray Nozzles

All Marley water cooling systems are equipped with Marley patented spray nozzles. More water now is cooled through Marley nozzles than through any other kind.

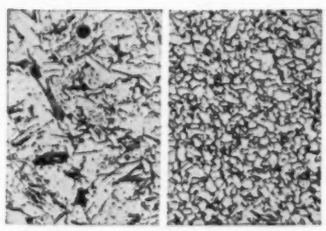
"Are your welds as STRONG as your claims? I'm from Missouri, Lad!"

Photo and data courtesy of The Ohio Fuel Gas Company, Columbus, Ohio

Technically Speaking



In welding with the Lincoln shielded arc, fusion takes place within a protective gaseous enclosure. The elements of the air cannot contact the molten metal; hence oxides and nitrides, which embrittle and weaken the usual weld, are excluded. The weld deposit is pure, homogeneous and in most cases of physical qualities superior to those of mild rolled steel.



These photomicrographs (enlargement, 100 diameters) show the grain structures of two welds. On the left is an ordinary weld; note the "nitride needles" and oxide pores. Contrast this to the weld on the right, made by the Lincoln shielded arc. Fine-grained welds like this are extremely strong, ductile and corrosion-resistant.

"Take a user's word for it, Pop! Here, EVERY ONE of 35 tests showed

'Shield-Arc' WELDS STRONGER THAN THE PIPE"

"Specimens of 20-inch, *'Shield-Arc'* welded pipe were crumpled by loads of 200,000 lbs.; the dents were then 'ironed out' by internal hydrostatic pressures of as high as 1820 lbs. per sq. in., causing 'blow-outs' in the pipe metal. 35 weld coupons from the various specimens were pulled apart in the laboratory. In every case, rupture took place outside of the weld. Invariably the results were tabulated, 'excellent,' '100%+.' The welds exceeded the user's requirements!

"When pipe joints like these can be produced so consistently . . . joints that are forever leakproof and more resistant to corrosion than the pipe itself . . . joints that *cost less* to make—is it any wonder that more and more *engineers are specifying 'Shield-Arc' Welding?* . . . Mail the coupon for valuable data . . . it's FREE!" THE LINCOLN ELECTRIC COMPANY, Department RR-250, Cleveland, Ohio. Largest Manufacturers of Arc Welding Equipment in the World.

THE LINCOLN ELECTRIC CO.
Dept. RR-250, Cleveland, Ohio

Please send me the following: ☐ "The Shielded Arc." ☐ A Guide to Lower Welding Costs. ☐ Data on *"SHIELD-ARC," the Complete Welder.*

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THE ONLY *complete* WELDER

CONSTANT IMPROVEMENT

by WAGNER

has resulted in

Quieter Air Conditioner Motors

WAGNER has furnished motors for air conditioning equipment for many years and has continuously improved motors for this purpose. Among the outstanding features of Wagner motors are: *(1)* QUIET OPERATION, *(2)* special operating characteristics to meet this exacting service, *(3)* long life, and *(4)* attractive appearance. As a result of Wagner's many years of experience in building motors for air conditioning equipment, Wagner motors are leaders in this field.

A few of the points of superiority of Wagner squirrel-cage motors are:

1. Rotor is of the cast-aluminum type, with bars, end-rings and blowers cast integrally, effecting a one-piece rotor exceptionally sturdy and long-lived. Skewed closed rotor slots minimize magnetic pulsations across the air gap and produce uniform starting torque at different positions of the rotor.

2. The frame and end-plates are strong and rigid, and accurately machined—assuring permanent alignment of bearings and uniform air gap, two essentials for quiet operation.

3. Blowers furnished on both ends of the rotor and many large openings in the stator frame insure thoro and unrestricted ventilation.

4. Stator windings are given a superior winding treatment which provides an insulating surface that is oil and moisture resisting and can be easily cleaned.

5. Stator and rotor laminations are punched from annealed, non-aging, high-grade electric sheet-steel selected for its magnetic properties.

Many other features of Wagner motors are worth investigating. Write today for Bulletin 174 which describes the Wagner motors that are Second-to-None for air conditioning.

Wagner Electric Corporation
6400 Plymouth Avenue, Saint Louis, U.S.A.

MOTORS • TRANSFORMERS • FANS • BRAKES

1303.8. **Recirculation and Air from Outside:** Provision may be made in the duct design for recirculation of air. Provided, however, that in such rooms where the occupancy by people is the governing factor not less than ten (10) cfm of air per person shall be introduced from the outside.

The inlets and ducts shall have sufficient capacity to permit the introduction of as much as 100 per cent of air from outside during the period of intermediate outside temperature subject to the approval of the Department of Buildings.

Section 1304. **Diffusion of Entering Air:** The location, shape, physical characteristics of, and air velocities and temperatures through the inlets and outlets for conditioned air to the rooms shall be such as will promote distribution of the circulation throughout all occupied portions without causing discomfort.

1304.1. Downward discharging air inlets in rooms shall be provided with safeguards to prevent drafts at the occupied zone. These safeguards shall include plaques, injector type grilles, or other devices which shall discharge the air at an angle to the vertical and which will prevent air colder than the room air from falling vertically downward at speeds capable of causing discomfort. Unless such safeguards shall be provided, the design entering air temperature from inlets discharging vertically downward shall be not more than 5 degrees colder than the room air at the approximate level of the inlet.

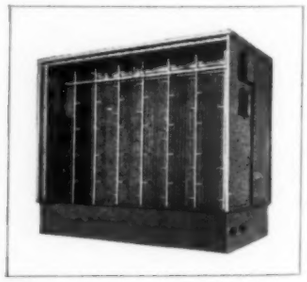
[*To be concluded*]

Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 126. Add the new products and companies listed here to your Directory Section which you received in your January 1936 HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

Single and Double-Stage Air Washers

No. 1000—Expanded and improved line of air washers includes single stage units built in two sizes (4 ft 10 in. and 6 ft 0 in. length of air travel) and double stage units in two styles: one (8 ft 0 in. in direction of air flow) has two spray chambers and one set of moisture eliminator plates and flooding nozzles on the outlet side of the second stage, and the other (9 ft 0 in. long in direction of air flow) has two spray chambers, two sets of eliminator plates and flooding nozzles and is in effect two single stage washers. Humidifying efficiency of the single stage washers is 65 per cent and of the double stage 85 per cent, states the maker.



Tanks are of heavy gauge steel, electrically arc welded throughout with all fittings for overflow and drain, spray and suction header, quick fill and float valve welded to the tank to make the washer one integral leakproof unit. Overflow and drain is combined into one unit for draining tank quickly for cleaning and flushing by simply unscrewing overflow pipe. Eliminator plates are of galvanized copper bearing steel and have six faces at 90 deg angles. All-brass centrifugal spray nozzles are of self cleaning type, will give complete atomization at pressures as low as 25 lb per sq in., at which each nozzle will pass 1.75 gpm, according to the manufacturer. Access doors, marine lights, and inlet distributor plates are provided.—National Fan & Blower Corp., 543 W. Washington St., Chicago, Ill.

You wouldn't do THIS to your shoes! . . .

. . . then why do THIS to your pipe lines?

• With a groove cut across the sole of your shoe every step would concentrate the strain at the weak spot. In a threaded pipe line all the stresses are concentrated at the weak points—THE THREADED PIPE ENDS—where 40% of the pipe wall has been cut away—where 40% or more of the effective thickness of your entire pipe system has been wasted.

THE pipe system weaves back and forth from expansion and contraction due to temperature changes. Vibration of machinery puts certain sections in constant motion. Sudden opening and closing of valves causes severe shocks in any pipe line.

There are no weak spots in pipe lines welded with WeldELLS at every turn and with Taylor Tees and other Taylor Welding Fittings throughout. None of the pipe wall is cut away. Nowhere is the pipe wall less than *full thickness* and *full strength*. At the crotch of the WeldELL, where the maximum stress occurs, the wall is made thicker, maintaining full strength. This reinforcing thickness does not encroach on the inside diameter. Thus in a WeldELL-welded pipe system there are no weak spots. The nominal pipe wall thickness is the effective thickness of the line.

Taylor Forge manufactures the only complete line of Seamless Welding Fittings.

40% of Pipe Wall is Cut Away in Threading!

This drawing was made from an actual sawed section of threaded 6" pipe. The black portion shows the amount of effective thickness destroyed by threading. The thread is only 1-5/8" long but the strength of the entire system against stresses and corrosion is penalized to the extent of the depth of the thread.

TAYLOR FORGE & PIPE WORKS, General Offices and Works, P. O. Box 485, Chicago, Ill. New York Office: 50 Church Street

TAYLOR FORGE

WeldELLS

and other WELDING FITTINGS

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Challenges
ANY BRAZING ALLOY

● For any copper or copper alloy brazing, Westinghouse stands back of this challenge! Phos-Copper has what it takes to produce high-quality brazed joints at low cost. Phos-Copper is the superior brazing alloy. It melts at a relatively low temperature, penetrates perfectly through tight joints, requires no flux on most applications, resists corrosion, possesses high ductility, high tensile strength and high fatigue resistance, and is economical and easy to use—characteristics that mean better joints at lower cost.

Notice how Phos-Copper flowed around the brass-to-brass plug joint at the bottom of this refrigerator dehydrator tank.

J 20110

Try PHOS-COPPER AT OUR EXPENSE

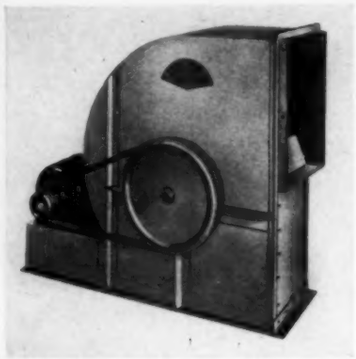
WRITE FOR A TRIAL SAMPLE. We will also send you, without obligation, a copy of Booklet 2076 which contains technical and test data on Phos-Copper. Address Room 5-N, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

WESTINGHOUSE 1886 W 1936 GOLDEN JUBILEE

Westinghouse

Unit V Belt Driven Blowers

No. 1001—New V belt drive unit blowers have been developed to meet the need for a combined unit to be used in place of a direct connected blower. The arrangement involves a "Uniblade Type H" blower operated by a standard frame motor operating at 1725 rpm equipped with suitable drive, the motor being mounted on a steel pedestal base built integral with the blower housing. Unit is compact, easy to install, and assures flexibility of control, necessary where it is required to increase or decrease air delivery, according to the manufacturer.



Five wheel diameters are available (12½, 15⅝, 18¾, 21⅞ and 25 in.), the largest with a capacity of 6345 cfm at 1 in. S.P. when driven by 2 hp motor.—Autovent Fan and Blower Co., 1805 N. Kostner Ave., Chicago, Ill.

Pressure Switch for Compressor and Pump Control

No. 1002—Features of new pressure switch for automatic control of small compressors and fluid pumps include compact all-metal enclosing case, flexibility of application, long, trouble-free operation; full automatic action, protection to motor, ease of wiring, adjusting and inspecting, according to the maker.

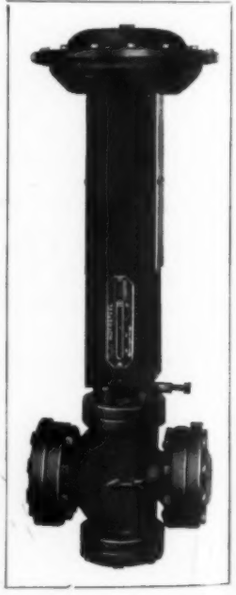
Pressure connection for S. A. E. fitting is furnished as standard, but may easily and quickly be converted into ¼ or ⅜ in. pipe fitting. Inverse time limit overload protection is obtained by thermal overload relay operating directly on the contact mechanism and giving protection to the motor during starting and running, allowing an ample time interval to take care of starting inrush and momentary overloads without tripping.

Split-type enclosing case and design of the mechanism permits easy access to terminals and adjustment screws.—Cutler-Hammer, Inc., N. 12th St. and W. St. Paul Ave., Milwaukee, Wis.

Diaphragm Valve Motors of All Steel Welded Construction

No. 1003—Redesigned line of "Moto-steel" diaphragm valve motors employs an all steel welded construction lighter in weight, stronger, and with greater resistance to shock and strains than older type. Other innovations include a smoother, quicker-acting, and more powerful valve motor obtained by a larger molded diaphragm of advanced design. Formed impressions in the top plate provide free access of air pressure to the diaphragm in starting to insure greater effective initial force. Longer and larger diameter spring gives greater initial compression, less hysteresis, and more accurate response, says the manufacturer.

All parts are specially treated to resist corrosion.—Taylor Instrument Companies, 95 Ames St., Rochester, N. Y.



RIC-WIL Gives You the Widest Choice of Designs in Underground Steam Conduit

FOR 25 years the name Ric-wiL has been identified with the best engineering practice on Underground Conduit for Steam Lines. Let us remind you that Ric-wiL ALSO offers the WIDEST range of designs and materials on the market. Illustrated above are only three of the several types of Ric-wiL Systems available.

With varied designs in both standard weight Tile and Super Tile, in Cast Iron, and in Armco Iron Unit Steam Main, with the famous Dry-paC Waterproof Asbestos Insulation, and other optional insulations and drainage methods—Ric-wiL is able to offer a comprehensive service adequate to ANY problem in the field of underground steam transmission.

Ric-wiL is THE dependable, perfected conduit that keeps your steam lines tight, dry, and 90% or more efficient. Ric-wiL Systems are complete, including installation instructions and engineering service drawings, also field supervision if desired. Find out what Ric-wiL has done for others! Submit YOUR heat or power transmission problems to Ric-wiL engineers! Catalog promptly on request.

UNDERDRAINAGE IS VITAL!

Heat loss from conduit in wet soil is over four times that in dry soil. Ric-wiL External Underdrain keeps the soil DRY.

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CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

for a **SUBSTANTIAL REDUCTION** *in* **YOUR WATER COSTS**

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WORTHINGTON pumping equipment of every type is available for each individual condition. These, with the advice of qualified Worthington engineers, assure the correct selection for any requirement.

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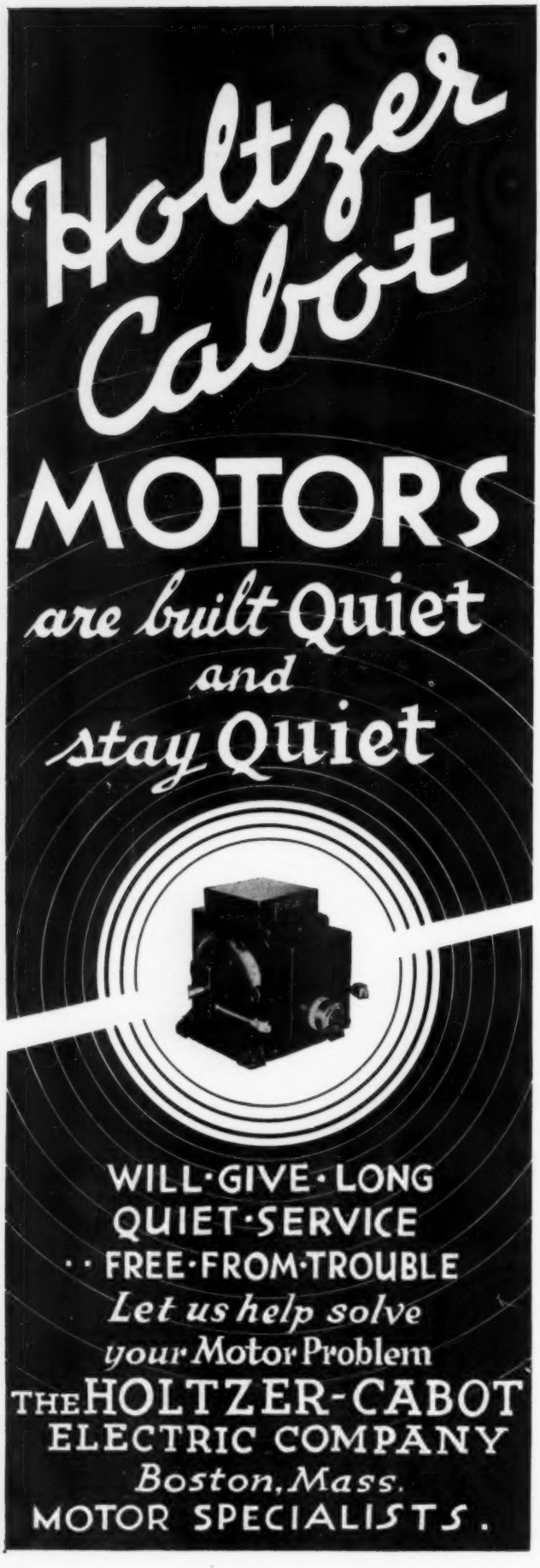


Combination Vise and Vise Stand

No. 1004—Combination vise and vise stand has vise taking pipe ⅛ to 2½ in.; stand includes pipe bender up to ¾ in., oil can recess and slots for wrenches, cutters or stock.

A choice of two pipe vises is offered, one with three jaws and the other with new type broad jaws which grip pipe along its entire length without marring. A feature of the new vise is the frictionless bearing in the form of a hardened steel disc which sets between screw and upper jaw, designed to give longer life to these parts and hold the pipe tighter without straining and effort.—Armstrong Mfg. Co., 303 Knowlton St., Bridgeport, Conn.

Motorized Speed Reducers

No. 1005—Two new sizes have been added to line of totally-enclosed, fan-cooled "MotoReduceRs" in both vertical and horizontal types.

Gear reduction is of planetary design and all gears and pinions are made of heat treated nickel steels. All gears and shafts are mounted on anti-friction bearings. Slow speed shafts are mounted

on tapered roller bearings having a high thrust rating. On the vertical types this construction allows vertical shafts, paddles, impellers, etc., to be hung directly from the driving shafts, eliminating step bearings. Horizontal types employ splash lubrication system, and the vertical units are equipped with a force-feed system. Both new types are water-proof and dust-proof.—Philadelphia Gear Works, Erie Ave. and G St., Philadelphia, Pa.

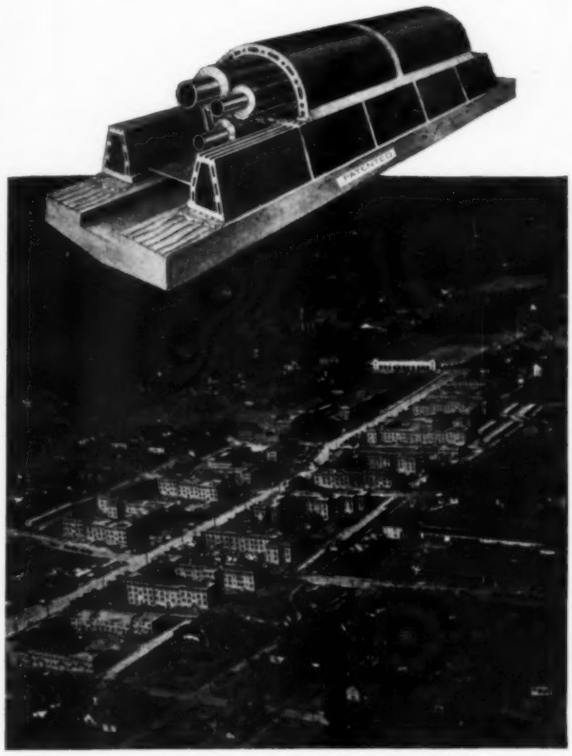
Indicating Temperature and Pressure Controllers

No. 1006—For processes where close throttling control is required but a record of the processing is not essential, an air-operated controller ("Fulscope") of the indicating type is offered. Features of this instrument, previously available only in

recording controllers, include either high range or full range sensitivity, as required, the former for processes having small time lags and the latter for time lags of any magnitude; control point adjustable to any value within the control range; controller action may be quickly reversed in the field by altering

THERM-O-TILE

STEAM CONDUIT



6,000 feet used for steam distribution at Federal Housing Project, Techwood, Atlanta, Ga. Covers 25 acres, supplying

Heat to 700 Families

Ask for Bulletin 351 describing complete details of this conduit system.

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Delaware — The Nicoly Corp., Philadelphia.
District of Columbia—Reid Hayden, Inc., Washington.
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Illinois—Asbestos & Magnesia Materials Co., Chicago.
Indiana—General Asbestos & Supply Co., Indianapolis.
Kansas — Johns-Manville Sales Corp., Kansas City, Kan.
Kentucky—General Asbestos & Supply Co., Louisville.
Louisiana — Taylor - Seidenbach, Inc., New Orleans.
Maryland—Reid Hayden, Inc., Baltimore.
Massachusetts — Springfield: Johnson Asbestos Co.
South Boston: F. S. Thorsen Co.
Mississippi — Taylor-Seidenbach, Inc., New Orleans.
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Pennsylvania—The Nicoly Corp., Philadelphia.
South Dakota — Johns-Manville Sales Corp., Omaha.
Texas — Houston: Aycock Corp. of Texas.
Dallas: Johns-Manville Sales Corp.
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Wisconsin—Milwaukee Insulation Co., Milwaukee.
Canada—Montreal and Toronto: Canadian Johns-Manville Co., Ltd.

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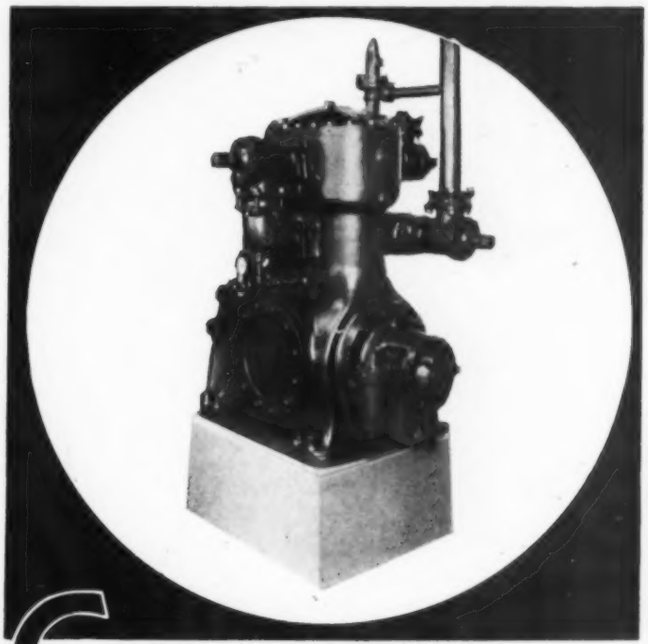
VILTER



COMPRESSORS

for "FREON"

The new Vilter "Freon" Compressor is the product of three years of intensive development by one of the country's foremost producers of industrial compressors.

The long years of successful experience in the field of refrigeration and cooling equips this company to render unusual service in handling individually engineered projects.

The compact size and proportions of the Vilter "Freon" Compressor make it ideal for any air conditioning application where floor space and head room are a factor and where "Freon" is to be used as the refrigerant.

Whatever your air conditioning requirements, consult with Vilter.

ACMA

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THE VILTER MANUFACTURING COMPANY
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2148 South First Street
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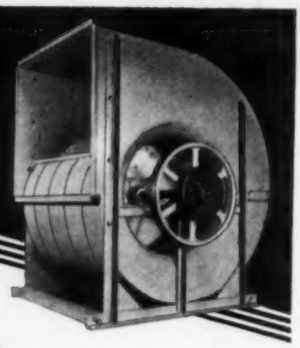

in Blowers it's the NATIONAL

Blowers of every size and type made to exactly answer every purpose—utility blowers, light duty, heavy duty, single and double inlet blowers—all expertly designed and ruggedly constructed to deliver at all times the maximum of satisfactory service.

No matter what air conditioning equipment you may need—blowers, ventilators, propeller fans, air washers, unit coolers, unit heaters, blast heat coils, exhausters, humidifiers, automatic shutters—you would profit by becoming familiar with the National line before making any purchase. Send for bulletins.

NATIONAL FAN AND BLOWER CORPORATION
543 W. Washington Blvd. Chicago, Ill.

BUILT FROM THE GROUND UP AND DOWN

The strength of the sturdy oak is not all above the ground. The oak depends upon its roots for life and growth. So with Pomona Pumps. It's not what you see above the ground that alone makes Pomona a great pump. Of course, it's recognized that there isn't a sturdier, better engineered pump head built, but that's only the half of it. It's the assembly of shafting, bearings, impellers and bowls that are hidden from passing gaze that are Pomona's roots. Here lies the secret of Pomona's superior performance, long life, and trouble-free operation. The high quality material, the meticulous machining, the precision fittings found in the bowl assembly of Pomona Pumps is seldom equalled, — has never been excelled. You are PAYING for Pomona performance. Be sure your pump is Pomona, — then you will GET IT.

Write for your copy of Pomona's new engineering circular on hydraulic laboratory pump testing.

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POMONA TURBINE PUMPS

the position of a link; many standard control ranges to choose from within the limits of minus 100 and plus 1200 F, or full vacuum and 3000 lb pressure; control ranges are interchangeable; temperature controllers may be equipped with mercury-, vapor-, or gas-actuated tube systems.

Large scale with bold numerals and graduations enable accurate observations from a distance. The black finished, cast aluminum case is available in both face and flush mounting styles.—Taylor Instrument Companies, 95 Ames St., Rochester, N. Y.

Welding Generator Embodies New Principle

No. 1007—Based on newly developed cross-field principle a new welding generator for belted or coupled welding sets with a novel control system said to produce superior arc quality and eliminating rheostat, resistors, meter, reactor and exciter required in conventional designs, has been announced. Due to its design the welding current is unaffected by the ordinary speed variations of the driving unit.

The control, which allows the operator to adjust the generator to give the welding current required before the set is started, is accomplished by mechanical means which vary the armature reaction fluxes of the generator. This control is continuous, without steps, and permits minute changes in current adjustment, states the maker. A polarity reversing switch is provided for instantly changing the polarity of the electrode as required by different electrodes without changing the cable connections. Elimination of the exciter makes it possible to use a short and heavy shaft to minimize vibration.

Following are pertinent specifications:

1 hr 50C rating amperes	Current range amperes	Speed rpm	Approx. Net Wt., Lb.	Approx. dimensions, in. length	width	height
200	60-250	1500	800	36	24	23
300	90-375	1500	1100	40	24	23
400	120-500	1500	1500	39	28	26

—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

New Steel Combines High Temperature Strength, Oxidation Resistance

No. 1008—New steel ("Silmo") has been designed for applications where an economical combination of high temperature strength and oxidation resistance is required; in the 1000-2000 F range, it has about twice the strength of carbon steel and its oxidation resistance compares favorably with that of 4-6 per cent Cr-Mo steel, states the maker. Indicated uses are for cracking furnace tubes, pipe still heater tubes, high pressure boiler and superheater tubes, and tubing used in the manufacture of air heating equipment.

Chemical composition is: carbon, 0.15 max.; manganese, 0.30 max.; phosphorus, 0.04 max.; sulphur, 0.045 max.; silicon, 1.15-1.65; molybdenum, 0.40-0.60. Complete physical data covering high temperature tensile properties, Charpy value at elevated temperatures, heat embrittlement tests, creep strength, temperature-stress embrittlement tests, stability, and corrosion and oxidation resistance, together with curves showing the creep strength at various temperatures and the relation between its creep strength and rate of creep at 800, 1000 and 1200 F are available. —The Timken Steel & Tube Company, Canton, Ohio.

De-Ionizing Motor Starters

No. 1009—New line starters employing the "de-ion" principle of arc interruption are designed for across-the-line starting for single phase and polyphase squirrel cage induction motors and as primary switches for wound rotor induction motors, are especially applicable where push button control is desired with complete protection to the operator, the motor and the machine.

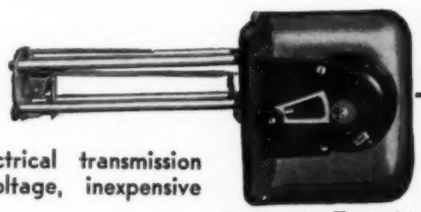
Motor protection is provided by a thermostat disc overload relay. An inverse time characteristic prevents shut downs due

FRIEZ BALTIMORE

Remote HUMIDITY Direct

For the first time in instrument history Relative Humidity is recorded directly from a remote location. The first reading is the last answer, without computation, without tables, without servicing.

The Butchers, the Bakers, the Candlestickmakers, Horticulturalists, Poultry Plants, Textile Factories, Department Stores, Banks, Public Buildings, are installing them as fast as they can be built—Direct Recorders of Remote Air Conditions, exclusively by Friez of Baltimore.



Positive, electrical transmission over low voltage, inexpensive wiring.

Temperature Transmitter

Superintendents . . . Executives . . . Customers read at a glance the complete current story of air-conditions in any department.

Bulletin R for Details Relative Humidity Transmitter

JULIEN P. FRIEZ & SONS, INC.

"The Makers of America's Weather Instruments"

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FEDDERS COILS

Manifolding Engineered for Effective Refrigerant Distribution

U. S. Pat. 2,023,749

The performance of an air conditioning coil depends on the engineering that goes into it. Fedders specially designed and patented manifolding provides distribution of refrigerant in accordance with the varying temperature differentials between air and refrigerant as the air passes through each section of the coil. Accurate refrigerant control by Fedders High Capacity Thermostatic Expansion Valves, provides a complete Low Side balanced in performance and assuring maximum effectiveness and current economy. Fedders factory-built manifolding also eliminates the hazards of field assembly.

Fedders Air Conditioning Coils are available in a complete range of capacities for use with refrigerant, cold water or steam. Write for Bulletin 91-A covering cooling and dehumidifying coils and Bulletin 101 covering heating surface.

FEDDERS *High Capacity* THERMOSTATIC VALVE

Easy, sensitive adjustment helps get greatest effectiveness out of each section of the coil. U. S. Pats. 1,974,631; 1,987,948.

FEDDERS MFG. CO.

57 Tonawanda St. Buffalo, N. Y.

AMERICAN-MARSH

PUMPS for Every Requirement

The quality of American-Marsh Pumps plus the experience of our engineering staff can help materially in reducing your pumping costs. We offer you expert knowledge gained through more than sixty years designing and manufacturing but ONE PRODUCT—PUMPS. And we build a complete line—CENTRIFUGAL, STEAM AND POWER DRIVEN PUMPS—modern in every way. You can profit by having our nearest representative call. Or write for Bulletins.

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BATTLE CREEK, MICHIGAN
PUMPS AND PUMPS ONLY SINCE 1873

NEW
PORTABLE UNISHEAR

CUTS UP TO
16 U.S. GAUGE
1/16" STEEL

No. 16

A definite need has been expressed for this portable Unishear of intermediate capacity. It is ideal for use where the requirements are too severe for the "Mighty Midget" Unishear with its capacity of 18 U. S. Gauge Steel and not sufficiently heavy to justify the purchase of a No. 144 Unishear with its capacity of 14 U. S. Gauge.

Patterned after the "Mighty Midget," the No. 16 is surprisingly light in weight, sturdily built and easy to handle. It is 100% safe.

Cuts straight lines, angles, notches and curves to a minimum radius of 1½". Speed up to 15 feet per minute.

Because of its large reserve capacity the No. 16 Unishear can be operated continuously as a production tool without ill effects.

Descriptive literature gladly furnished

STANLEY ELECTRIC TOOL DIVISION
THE STANLEY WORKS
139 Elm St., New Britain, Conn.

STANLEY UNISHEARS
MODELS AVAILABLE WITH CAPACITIES UP TO ¼" BOILER PLATE

to temporary overloads, and protection is assured against single phase and locked rotor operation. Heaters for operating the disc are interchangeable, easily accessible. The relay can be arranged for either manual or automatic reset.

The "de-ion" arc quencher confines, divides and extinguishes the arc almost instantaneously and without the usual flash and

attendant scattering of the flame, giving not only high arc rupturing ability but also assuring long life of the contacts, states the maker.—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Recorder—Controllers with Wide Range Sensitivity Adjustment

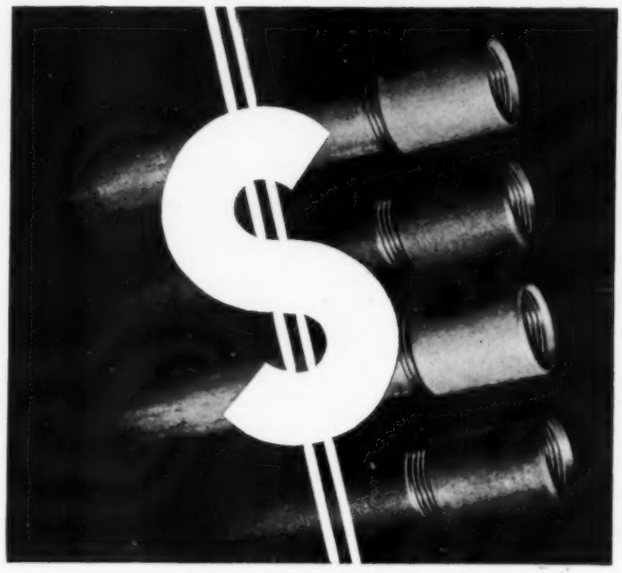
No. 1010—New series of pneumatic type controllers known as "Ampliset" free vane controllers are available for automatically controlling temperature, time-temperature, flow, liquid level, pressure, time-pressure and humidity. The "Ampliset" principle has also been combined with the "Metameter," a system of telemetering.

In the new controller features of field adjustability have been incorporated whereby its sensitivity can be changed by means of a small screw driver to fit the lag peculiarities of the process on which it is used. The sensitivity or throttling range of the instrument can be changed without removing the chart or chart dial and without disturbing the operation of the controller or upsetting the operation or equilibrium of the process. By the same adjustment that regulates the sensitivity, the instrument can be changed from direct-acting to reverse-acting, or *vice versa*, simply by turning the scale on the adjustment past the zero point.—Bristol Co., Waterbury, Conn.

Flexible Coupling for Fractional Horsepower Drives

No. 1011**—Flexible shaft coupling for fractional horsepower power transmission utilizes oil resistant rubber cushioned compression drive principle enabling unit to operate efficiently under conditions of parallel or angular misalignment, according to the maker. Coupling is made up of three simple members.—Certified Flexible Couplings, Inc., 122 E. 42nd St., New York, N. Y.

A MATTER OF



and sense

To obtain contracts these days, it takes close figuring on bids. Here's how Fretz-Moon Pipe can help you install quality work at a profit. • Fretz-Moon Pipe is absolutely uniform. Every length is true to size, perfectly round, straight, accurately and cleanly threaded, tightly welded, and clean inside and outside. It bends, cuts and threads easily—the exclusive "continuous process" by which Fretz-Moon Pipe is made insures against hard or burnt spots in the metal—protects the original high ductility of the steel. It is available in extra long lengths without cross-weld. In galvanized finish, it affords a rust-resisting zinc alloy surface that will not crack or scale under severe bending. • And with all these features that make fitting easier and faster, and cut installation time to a minimum, Fretz-Moon Pipe costs no more than ordinary pipe. Use it for your next piping job and note the difference.

FRETZ-MOON TUBE CO., INC.
BUTLER • PENNA.

FRETZ-MOON
PIPE
BLACK AND GALVANIZED

Corinco Cork Pipe Covering Remains Efficient

One of the important questions to be asked BEFORE drawing specifications for installation or replacement on a refrigeration plant is: How long will the required pipe covering retain its calculated efficiency?

Corinco Cork Pipe Covering for cold lines offers a degree of permanence in protection against refrigeration losses which makes it the outstanding insulating material in any plant where low temperature insulation plays a part. Corinco Cork Pipe Covering and fittings are moulded and the method of manufacture and application to the installation is so exact that attention or special care is practically eliminated.

Permit us to furnish facts to prove it to you.

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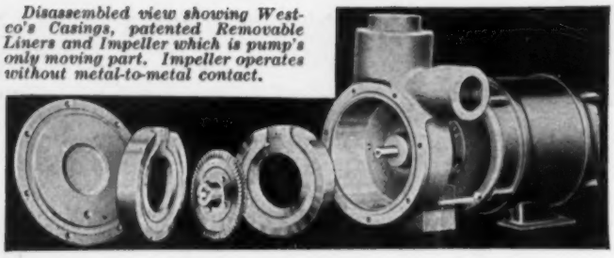
Disassembled view showing Westco's Casings, patented Removable Liners and Impeller which is pump's only moving part. Impeller operates without metal-to-metal contact.



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PREVENT costly circulation failures by specifying or installing WESTCO TURBINE PUMPS. Because a Westco's capacity is NOT materially affected by considerable increase or decrease in head.

Horse power consumption decreases in proportion to any head decrease, thereby maintaining efficiency and guarding against motor overload.

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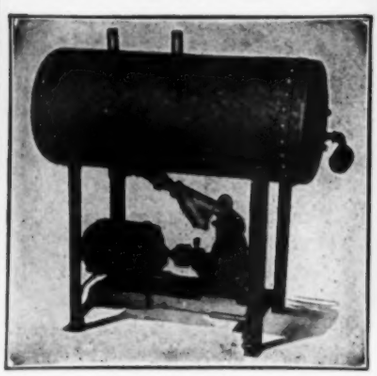

Designed and built by makers of automatic gas boilers for over a quarter century, the M. K. O. Boiler Feed *automatically* pumps feed water against high boiler pressure . . . returns condensate to boiler . . . supplies make-up water as needed. Used with gas, oil, stoker or hand fired boilers, it holds down steam costs . . . saves many times its price.
Write today for Bulletin 4-B, giving details of its construction and performance qualities. It will pay you.

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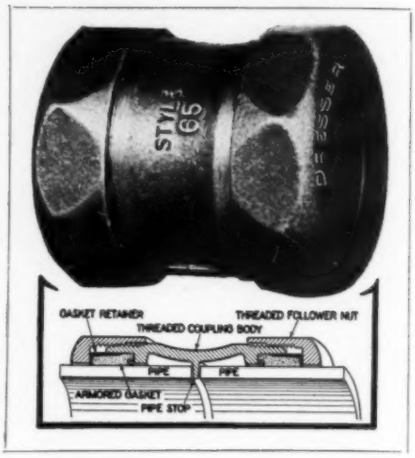
Compression Fittings for Plain End Pipe

No. 1012—Compression fittings, known as "Style 65," have been developed for making tight, flexible connections on plain end pipe, using only a wrench. Two threaded "follower" nuts, when screwed up on the threaded body of the fitting, compress two resilient, rubber compound gaskets around the pipe, sealing in contents of line. Fittings come assembled ready for use.

Conforming to specifications for 150 lb malleable fittings, they may be used on standard steel pipe 2 in. I. D. or smaller for gas, water, oil, compressed air, process and other lines. Gaskets retain their resilience except where temperatures exceed 200 F, states the maker.



Various types are available, including standard length body, extra long body, 45 deg and 90 deg ells, tees, adapters, etc. Advantages stressed are flexibility to absorb pipe movement, easy of making repairs and cut ins, possibility of re-use where temporary piping is installed or lines are periodically taken up and relocated.—S. R. Dresser Mfg. Co., 41 Fisher Ave., Bradford, Pa.

Window Ventilator and Filter Unit

No. 1013—New model of "Filteraire" unit has been designed to be modern in appearance without being extreme. Rated capacity is 450 cfm. Unit comprises replaceable type "Airplex"

air filter, centrifugal blower, and acoustical chamber or silencer. Furnished either with a-c or d-c variable speed motor.—Davies Air Filter Corp., 390 Fourth Ave., New York, N. Y.

Pipe Line Flow Signaling Device

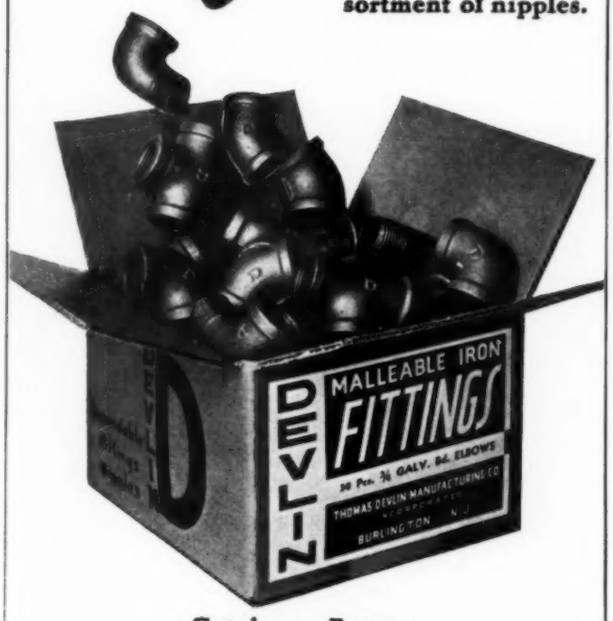
No. 1014—Swinging vane type of force fluid-metering device, designed for alarm purposes as in connection with sprinklers, and hence provided with electric contacts but not with an indicator and scale consists of (a) housing with its contact equipment, (b) flexible paddle suspended from the electrical unit, and (c) strap to hold the housing in assembly over a hole cut in the pipe line.

The paddle, of flexible material, can be inserted in a hole considerably smaller than its extended diameter. An alarm can be sounded on a sprinkler line in the event of fire or accidental sprinkler operation. Pumps, valves, motors, ventilators can be made to operate on starting, stopping or fluctuating flow. An electrical retard can be specified to prevent operation resulting from momentary surges, hammer or sudden pressure changes. Electrical units are mounted on the housing under a molded glass cover. Parts subject to corrosion are suitably protected.—The Autocall Co., Shelby, Ohio.

More and more Contractors and Maintenance Engineers are ordering Devlin Fittings and Pretz Nipples in cartons for greater convenience and to save handling.

21 different cartons of fittings.

5 different carton assortment of nipples.



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Air Conditioning

The distribution of air through ducts at speeds, and in quantities to meet various conditions, presents no great problem to you and your engineering associates. Given proper equipment, the control of the temperature, and humidity of such air currents is a simple matter.

THE BASIS IS ELECTRIC REFRIGERATION

Equipment is the basic factor. Summer air conditioning depends for its effectiveness on skill that has gone into the design, manufacture and application of the electric refrigeration equipment, *which makes the whole system possible.* Such air conditioning is basically taking the excess heat and moisture out of the air. The accepted method of doing this is by electric refrigeration. So we believe that when you consider air conditioning you cannot do better than to look first of all for the leader in electric refrigeration. You do not have to look far. Delco-Frigidaire, with its General Motors background, represents such experience to an unparalleled degree.

We welcome opportunities to discuss air conditioning problems either in person or by mail.

THE FACTS IN A NUTSHELL
2 ACCEPTED FACTS ABOUT AIR CONDITIONING THAT POINT TO ONE CONCLUSION
1 Summer air conditioning is basically cooling and dehumidifying.
2 The accepted method of accomplishing this is by electric refrigeration.
CONCLUSION
Buy air conditioning from Delco-Frigidaire—the organization representing most experience in electric refrigeration.

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See Sweet's Catalog File (Sec. 26/7) for air conditioning equipment details.

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Every Ilg installation gives you three profits, (1) the product—(2) labor—(3) materials. Ilg Electric Ventilators are made in a complete range of sizes from 12 to 72 inches. They are tested and rated in accordance with code of the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers. Send for 32-page book giving complete details on the full line of Ilg Electric Ventilators, Heating and Air Conditioning Equipment. Ask for specific information on any problem. Write now!

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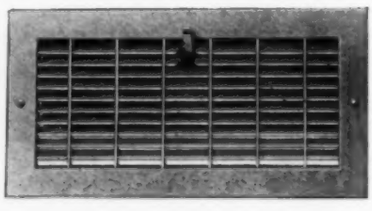

Tuttle & Bailey

The Editor's Correspondence

Efficiencies with Various Fuels

THE EDITOR—Kindly advise us of the efficiencies that might be expected with a horizontal return tubular steel boiler fired with a horizontal rotary oil burner; buckwheat coal hand fired; barley, stoker fired; buckwheat, stoker fired; buckwheat, forced draft, hand fired; barley coal, forced draft, hand fired.—C. G. S.

REPLY—The efficiencies would run approximately as follows:

Fuel oil, automatically fired with a horizontal rotary oil burner	75%-78%
Coal, buckwheat, hand fired	55%-65%
Barley, stoker fired	65%-72%
No. 1 Buckwheat, stoker fired	65%-72%
No. 1 Buckwheat, forced draft, hand fired	62%-70%
Barley coal, forced draft, hand fired	62%-70%

The figures necessarily cover a range in efficiencies in each case. The maximum efficiencies on this type of boiler would be obtained at between 80 and 120 per cent of boiler rating; both above and below this point the efficiencies would tend toward the lower figure.

In the case of hand firing the efficiencies would vary largely, depending on how closely the damper was watched, particularly in regard to maintaining correct overfire draft.

The efficiencies stated for burning fuel oil are based on using a modulating fire which correctly synchronizes the air admitted to the burner with the oil to maintain a correct steam pressure. It also assumes the use of an overfire draft controller to stabilize the draft with changes in outside wind and barometric conditions.

If the burner is to be operated intermittently, the overall efficiencies will probably be lower, approximately 5 per cent, due to the air infiltration while the burner is not running and to the fact that high combustion efficiencies will not be achieved until after a few minutes operation each time the burner starts. Part of the loss of efficiency with intermittent combustion can be eliminated with the use of an electric damper motor that closes completely each time the burner stops and opens to a wide open position before the burner can start again and then throttles to maintain correct overfire draft after ignition is complete.

An article to appear in an early issue of HEATING, PIPING AND AIR CONDITIONING will contain details on the auxiliary equipment necessary to maintain high efficiencies.—PAUL R. UNGER.*

*Mid-West Heat Service Co., Chicago, Ill.

Book Reviews

Figuring Air Conditioning

"How to Figure Air Conditioning," by Harold M. Hendrickson, Engineer, York Ice Machinery Corp. Published by the American Society of Refrigerating Engineers, 37 W. 39th St., New York, N. Y. 56 pp. + psychrometric chart and table of air-water vapor mixtures, 6 x 9 in., clothbound.

This compact book is a step-by-step explanation of the calculation of the sensible heat gain, air quantity, latent heat gain, and refrigeration load for the design of air conditioning systems, with information on use of the bypass, spray type dehumidifiers, surface cooling, evaporative cooling, and air conditioning engineering for high altitudes. Each subject considered is illustrated by calculations for a typical problem worked out in detail on the basis of the information given in the text, much of which is tabulated.

In his introduction, the author points out the differences in figuring heat *gains* for air conditioning design and heat *losses* for heating system design, selection of dry and wet bulb design temperatures, and inside design conditions and effective temperatures. Under sensible heat gain, solar heat is given careful treatment and deductions for shading explained, as are temperature differentials for calculating transmission heat gains through walls and roofs.

The text and sample calculations on evaporative cooling are

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Engineers and contractors are awake to the advantages of specifying Baker Freon Units in air conditioning applications both at home and abroad. In America, the demand for Baker Dual-mounted, Automatic Capacity Control Units keeps factories working night and day. And foreign engineers, long familiar with the rugged design of Baker compressors, find their requirements for dependability, economy and flexible cooling power more than satisfied by units from the Baker line. Baker builds 44 sizes, ranging from ¾ to 60 H. P. *especially for air conditioning service.* Write for data.

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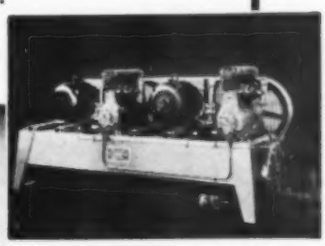
Engineering and Sales Everywhere

This beautiful Provincial home is air conditioned throughout with Baker equipment.
BURGE ICE MACHINE CO.
Engineers

Baker units deliver low cost cooling power for the complete air conditioning system of the Jockey Club Brasileiro, Rio De Janeiro, Brazil.
CEIBRASIL REPRESENTACOES, Ltda.
Designers and Engineers



Baker Dual Condensing Unit with Automatic Capacity Control—available in a wide range of capacities.

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LEVER ACTION

600% MORE POWER

Positive Control OF OIL OR STEAM AT HIGH PRESSURE

The GENERAL K10 lever action develops 6 times the opening and closing power of straight magnetic valves. Unfailing delivery and positive shut-off are assured on fluids or gases at high pressure. Approved by the Underwriters' Laboratories. Write for Catalog J-182.

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of particular interest, and the applications and limitations of this type of system are stated. In considering engineering for high altitudes, the effect of the lower barometric pressure on the standard psychrometric chart and tables of air properties is clearly explained.

Commercial Refrigeration

"Commercial Refrigeration," by L. Kay Wright, Refrigeration Instructor, Y. M. C. A. Trade and Technical School, New York City. Published by Nickerson & Collins Co., 435 N. Waller Ave., Chicago, Ill. 403 + x pp., 6 x 9 in., clothbound. Price, $3.50.

Covering the entire field of commercial refrigeration in the smaller installations, this treatise aims to present the necessary information and data required to undertake a job from the original design to the operation. Written to appeal to the "practical" engineer, the author has endeavored to present his material as devoid as possible of theoretical formulas and lengthy calculations, basing his discussion for the most part on actual installations.

Subjects covered include preservation of foods, refrigeration fundamentals, estimation of refrigerator problems, equipment, heat loads and coilage determinations, applications in bakeries, biscuit manufacture, drinking water systems, dairies and creameries, ice cream, and display cases. Two chapters are devoted to methyl chloride, sulphur dioxide and freon applications and data; in the main the book pertains to ammonia, as this refrigerant finds wide use in the small unit commercial field.

Two chapters are given to a rather general treatment of air conditioning, stress naturally being placed on refrigeration requirements.

Oil Heating Handbook

"Oil Heating Handbook," by Han A. Kunitz, Consulting Engineer. Published by J. B. Lippincott Co., East Washington Square, Philadelphia, Pa. 456 + viii pp., 5 x 7½ in., bound in flexible cloth. Price, $3.50.

According to the jacket, this is "the all-inclusive guide for every man who designs, installs, sells or uses oil heating equipment" and perhaps it is from the attempt to make this volume "all-inclusive" that it suffers most. As a result, much material is included which is more suitably and ably treated in other texts and handbooks and the treatment of those subjects which one would expect to find in an oil heating handbook is impaired. For instance, but six pages are devoted to oil burners themselves, while a 16 page section on terminology includes definitions of such terms as "barter," "bill of lading," "F. O. B.," "circumference," and so on.

Part I (10 pp.) is devoted to oil burner systems and classifications; Part II (129 pp.) to "combustion," including information on geology and oil surveying; Part III (118 pp.) to various types of heating systems and controls; Part IV (52 pp.) to surveys and installations, including a general discussion of sound and noise; Part V (27 pp.) to retail selling, commandments for salesmen, etc.; and Part VI to forms, tables, and terminology. There are numerous illustrations, an oil burner service chart, a sample contract, and other forms of direct and indirect interest.

Conventions and Expositions

Stoker Manufacturers' Association: Annual meeting, June 11-13, Greenbrier Inn, White Sulphur Springs, W. Va. Headquarters Office, 307 N. Michigan Ave., Chicago, Ill.

American Society of Mechanical Engineers: Semi-annual meeting, June 15-20, Hotel Adolphus, Dallas, Texas. Secretary, C. E. Davies, 29 W. 39th St., New York, N. Y.

National District Heating Association: Convention and exhibit, June 16-19, Pantlind Hotel, Grand Rapids, Mich. Acting

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A complete line of equipment for heating and cooling is at your service. The home office in Newark or any of our branch offices will gladly send complete descriptive literature or render prompt personal and efficient technical co-operation. Simply write to the address below.

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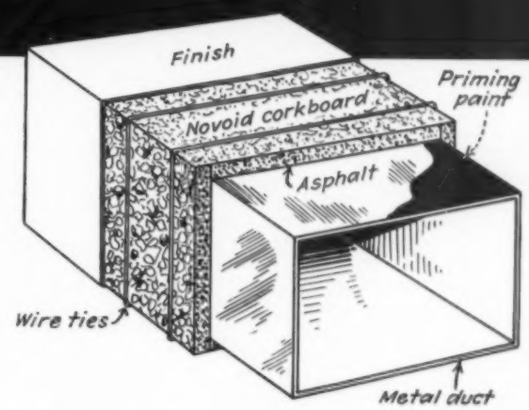

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This efficient corkboard is a natural and impassable barrier to heat, cold and moisture. And it's a fire-retardant, too. It is structurally strong, light in weight and may be easily and quickly erected on square or rectangular ducts. Once properly installed, Novoid Corkboard will cut operating costs and give your customers many years of satisfactory and economical service. Send the coupon for complete information including detailed drawings showing the simple application of Novoid Corkboard.

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Motor Entirely Isolated From Air Stream . . . .

Wherever hot greasy air, acid laden air, corrosive fumes or heavy dust must be conveyed . . . yet isolated from the motor . . . the De Bothezat BIFURCATOR is invaluable. Can be installed in any ventilating system without necessitating change in direction of air flow or size of main line. De Bothezat BIFURCATORS form a part of the duct system and are easily mounted. They employ the famous De Bothezat disc pressure fans . . . in sizes from 8" to 48" . . . and are built to meet the specific application: constructed of metals resistant to the gases being exhausted. Write for descriptive circular.

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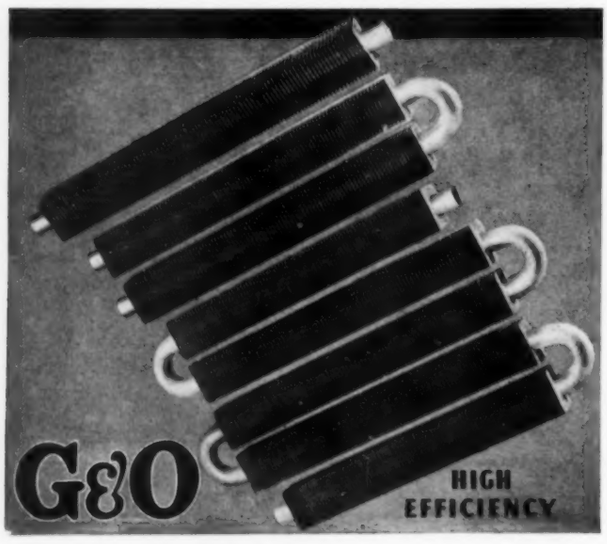

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Special Shapes, Bends and Continuous Coils

Correspondence invited.

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Pioneer Manufacturers of Individual Square Fin Tubing in the United States

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National Warm Air Heating and Air Conditioning Association: Convention, June 17-18, Deshler-Wallick Hotel, Columbus, Ohio. Managing Director, A. W. Williams, 50 W. Broad St., Columbus, Ohio.

International Congress of Refrigeration: June 16-27, The Hague and Amsterdam, Holland. Organization Office, 107 Stolberglaan, The Hague (Holland).

American Society of Heating and Ventilating Engineers: Joint semi-annual meeting with *American Society of Refrigerating Engineers,* June 22-24, Buck Hill Falls, Pa. Secretary, A. V. Hutchinson, 51 Madison Ave., New York, N. Y.

Short Course in Coal Utilization: June 9-11, University of Illinois, Urbana, Ill. A. C. Callen, Head of the Department of Mining and Metallurgical Engineering, 209 Transportation Bldg., Urbana, Ill.

American Society for Testing Materials: Annual meeting June 29-July 3, Chalfonte-Haddon Hall, Atlantic City, N. J. Assistant Secretary, R. E. Hess, 260 S. Broad St., Philadelphia, Pa.

Engineers' Day, Great Lakes Exposition: July 11, Cleveland, Ohio. General Chairman, Great Lakes Exposition, Dudley S. Blossom, Terminal Tower, Cleveland, Ohio.

National Association of Power Engineers: Annual convention and exhibit, August 31-September 4, Hotel Stevens, Chicago, Ill. Secretary, F. W. Raven, 176 W. Adams St., Chicago, Ill.

National Power Show: November 30-December 5, Grand Central Palace, New York, N. Y. Manager, Charles F. Roth, Grand Central Palace, New York, N. Y.

American Society of Mechanical Engineers: Annual meeting, November 30-December 5, New York, N. Y. Secretary, C. E. Davies. 29 W. 39th St., New York, N. Y.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 126

No. 2000—ACOUSTICAL MATERIALS: Burgess Battery Co., 111 W. Monroe St., Chicago, Ill. (a) 4 p. bulletin describing "Acousti-Pad" noise absorbent for lining air conditioning ducts, comprising a blanket of balsam wool with a perforated metal facing, giving chart for determining duct length to be lined, information on noise levels and a chart showing head restriction due to lining existing ducts; (b) 4 p. bulletin describing in detail methods of installing "Acousti-Pad," including information on calculating splitters and duct sections; (c) 4 p. bulletin discussing noise and how it is measured, including decibel calculation problems.

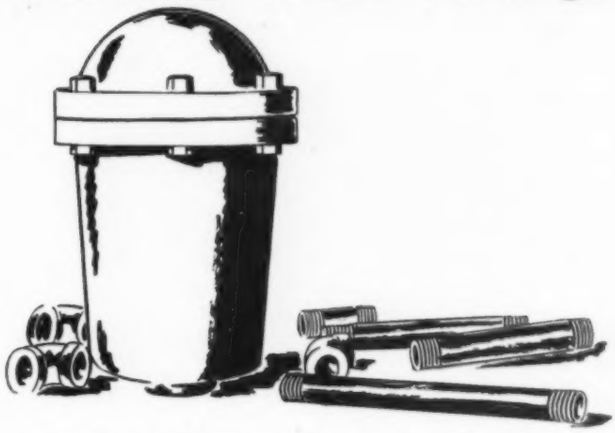
No. 2001—AIR CIRCULATORS: Air Controls, Inc., 1933 W. 114th St., Cleveland, O. 4 p. bulletin on "Rex-Airate" large volume air circulators with floor legs or for ceiling suspension, with information on selection and specifications.

No. 2002—AIR CONDITIONING: Airtemp, Inc., 8021 Conant Rd., Detroit, Mich. (a) 18 p. bulletin stressing the importance of air conditioning and its application to offices, stores, hotels, theaters, hospitals and other buildings, and showing jobs using circulating water, direct expansion, ice bunkers, deep wells and other types; (b) data sheet on self-contained air conditioners, with ¾ hp compressor motor, handling 200 cfm; (c) 4 p. bulletin on suspended type air conditioners designed for automatic air bypass control as an optional feature for handling air directly into conditioned area or to be used in connection with duct systems; (d) 4 p. bulletin on compressors, sizes ¾ to 25 hp.

No. 2003—AIR CONDITIONING: Baker Ice Machine Co., Inc., 1509 Evans St., Omaha, Nebr. 20 p. spirally bound booklet

WHICH WILL IT BE?

A Complete Steam Trap..or..an Ordinary Trap and Buy These Fittings



The illustration above shows Super-Silvertop, the only complete steam trap on the market. Super-Silvertop is installed straight-in-line or as an elbow; therefore it requires no elbows or nipples. Elbow facilities are cast in the head of the trap, eliminating elbow connections in either direction. The trap handles ordinary dirt and scale—hence no strainer is needed. Super-Silvertop works on a vacuum, proof that it does not leak steam. Together with these features, you can have a snap action Thermal Air Eliminator that assures instant heat to the unit. All these advantages are yours in one steam trap.

Now compare Super-Silvertop with the trap at the right. In the case of this trap you need, in most installations, three elbows and four nipples. Not only that, this trap cannot have snap action thermal air elimination. It takes from 40 to 60 minutes longer to install this trap than the one at the left. Which one will you buy—the trap that requires all these extras or Super-Silvertop, the only complete steam trap?

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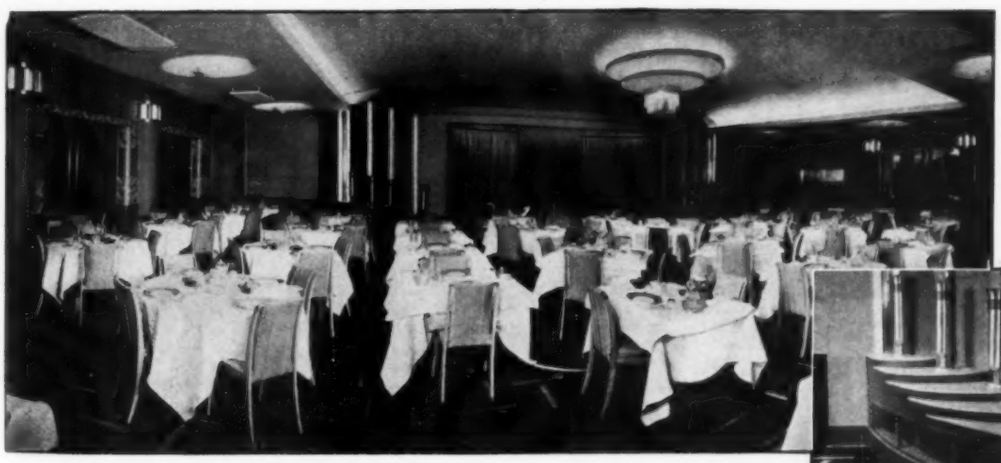

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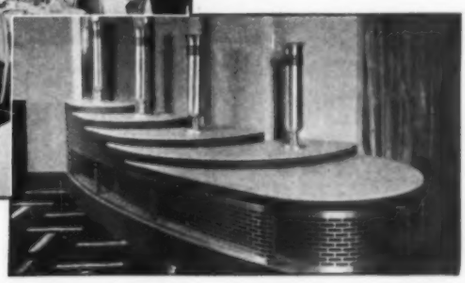

Another notable instance of the fact that AUER makes types of Registers or Grilles suitable for every decorative scheme. Complete Catalog on request.

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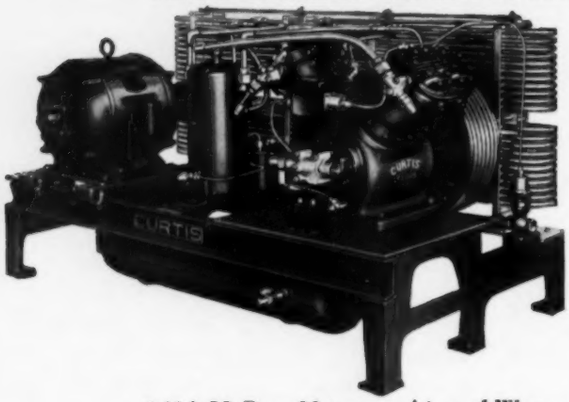
AIR-CONDITIONING UNITS AND COMMERCIAL REFRIGERATION



1/6th H. P. to 30 tons—Air and Water cooled. Extra capacity—Slow operating speed—Experienced design—Low up-keep—Rugged construction—Fine materials and Precision workmanship.

Some desirable territories still open.

Advanced design, soundly engineered. 82 successful years—financial stability.

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HERE'S WHAT PROTECTS PROFITS
True-ground, bronze ball-joint stays tight no matter how often removed and remade.

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For a Dart joint stays tight, no matter how often removed and reinstalled. The two wide seats of special bronze alloy, ground on special oscillating grinders to a true ball joint, are easy to take down and remake whenever piping is changed. And Dart bodies are heavy, durable. Breakage and stretch are practically eliminated. It takes extra operations to make unions like this—which means extra service for you, at less cost. Test Dart economy for yourself. Tell us what size you use most. We'll gladly send you one.

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E. M. DART MFG. CO., PROVIDENCE, R. I.
Sales Agents: The Fairbanks Company, New York, and all branches. Canadian Factory: Dart Union Company, Ltd., Toronto, Canada

on commercial and residential air conditioning with information on advantages of air conditioning stores, shops, restaurants, theaters, radio studios, public buildings, storage rooms, funeral homes, clubs, hotels and homes, and illustrating this company's air conditioning and comfort cooling equipment.

No. 2004—ATOMIZING UNITS: Blower Application Co., 918 N. Fourth St., Milwaukee, Wis. 4 p. bulletin on "Cycloidal" atomizing units and air washers for industrial air conditioning applications, particularly in malt house attemperator rooms, giving table of capacities, dimensions and other data. Units consist of fan cooled motor with shaft extension having cycloidal atomizing wheels mounted thereon.

No. 2005—BOILERS: Kewanee Boiler Corp., Kewanee, Ill. 14 p. catalog of "Type C Electric-Weld" steel boilers, giving complete information on design and construction characteristics, specifications for oil, gas or stoker firing, dimension drawings, etc.

No. 2006—COMPRESSORS AND VACUUM PUMPS: Ingersoll-Rand Co., 11 Broadway, New York, N. Y. 28 p. catalog covering line of industrial compressors and vacuum pumps; compressors single and two stage, ¼ to 15 hp, 1.2 to 102 cfm, pressures to 1000 lb, vacuums to within 0.35 in. of mercury.

No. 2007—CONDENSATE RETURN UNITS: Worthington Pump & Machinery Corp., Harrison, N. J. 4 p. bulletin on "Monobloc" condensate return units for building and plant service with dimension drawings, tables and ratings. Units for from 2000 through 15,000 sq ft radiation have vertical welded steel tank and from 20,000 through 40,000 sq ft radiation, horizontal tank.

No. 2008—CONVECTORS: Young Radiator Co., Racine, Wis. 4 p. leaflet on "Streamaire" convectors, unit heaters, blast heaters, heat transfer surfaces and unit coolers, illustrating and describing the equipment and discussing advantages and applications.

No. 2009—DUST AND FUME COLLECTION: Claude B. Schneible Co., 4755 N. Pulaski, Chicago, Ill. 24 p. catalog illustrating and describing wet method of dust and fume collection in foundries, food products, chemical and other industrial plants, describing the equipment, giving capacities and dimensions of multi-wash collectors and showing installations.

No. 2010—ELECTRIC HEATING: Hynes Electric Heating Co., 240 Cherry St., Philadelphia, Pa. 16 p. bulletin on industrial electric heating systems with automatic control, including heaters for circulating oils or other fluids, electric heating of industrial vessels and processes, pitch and compounds, pipe coating, precision bearings, etc., electric steam and hot water boiler circulation heaters, ice melting heaters and tubular heaters. Features of the equipment and applications are described.

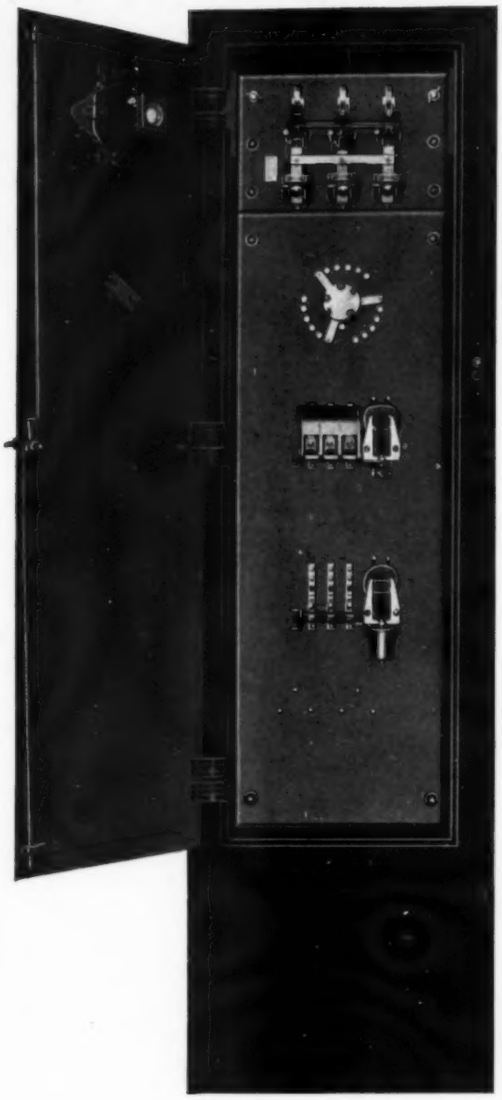
No. 2011—ELECTRIC STEAM BOILERS: Commonwealth Electric & Mfg. Co., 83-105 Boston St., Boston, Mass. Data sheet on electric heated, full automatic steam boilers, capacities 3 to 25 boiler hp, pressure ranges 0-100, 0-200 and 0-300 lb, with price lists and specifications.

No. 2012—FANS AND BLOWERS: Autovent Fan & Blower Co., 1805 N. Kostner Ave., Chicago, Ill. Looseleaf catalog on centrifugal and propeller fans, blowers and steam and electric unit heaters, giving information on construction, capacity tables and dimension data, with data sheets on automatic louvres and accessory equipment.

No. 2013—GEARS: The Ohio Gear Co., 1333 E. 179th St., Cleveland, O. 128 p. catalog describing complete line of spur, bevel, worm and other gears and motorized speed reducers, with technical data on gearing, standard S.A.E. heat treatment methods and other pertinent data.

No. 2014—HOT WATER HEATING: Erwin L. Weber, 534 Medical Arts Bldg., Seattle, Wash. 4 p. bulletin of hot water heating data announcing a new service, the balancing of systems by means of the electro-differential thermometer when using equalizing non-reflux elbows, a patented development for balancing forced circulation hot water heating systems no matter how improperly proportioned the piping system may be

Clark-Sundh



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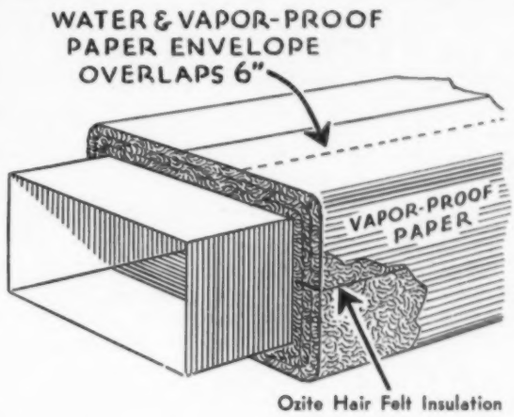

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FOR ALL SOUND, HEAT AND COLD INSULATION PURPOSES

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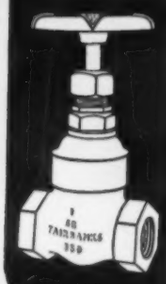

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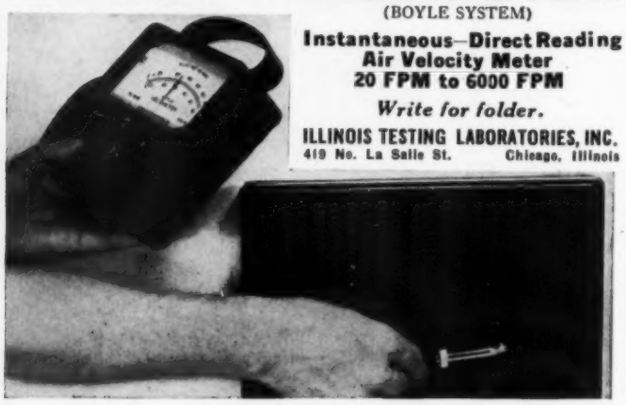

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20 FPM to 6000 FPM

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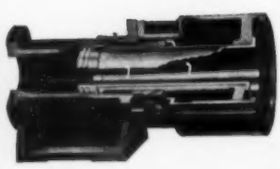

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Never fails from corrosion produced by electrolytic action. It is all-cast aluminum and iron radiation—*no other unit heater like it.* Built to last indefinitely without maintenance. Send for details now.

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designed or installed. Notes on estimating pipe sizes are given.

No. 2015—INSTRUMENTS: General Instruments & Controls Co., 205 W. Wacker Dr., Chicago, Ill. 4 p. folder on instruments for air conditioning, including humidistats, thermostats, windowstats, insertion 'stats, portable recorders, humidity indicators, anemometers, direct reading air velocity meters, psychrometers and others.

No. 2016—INSTRUMENTS: Practical Instrument Co., 2717 N. Ashland Ave., Chicago, Ill. Leaflet describing 24 hr recording thermometer and 24 hr electrical operation recorder, with information on features, uses and list prices.

No. 2017—MOTORS: The Lincoln Electric Co., 12818 Coit Rd., Cleveland, O. 6 p. bulletin on facts for users of a-c motors, listing points to check when buying motors and giving data on general purpose squirrel cage induction motors.

No. 2018—OIL BURNERS: National Airoil Burner Co., Inc., 1327 Girard Ave., Philadelphia, Pa. 4 p. bulletin describing and giving specifications for automatic oil burners for apartment houses, churches, stores, business buildings and small factories, with capacity and dimension tables.

No. 2019—PIPE: Republic Steel Corp., Republic Bldg., Cleveland, O. 64 p. booklet on "Toncan" iron pipe, illustrated with many test charts and tables, photographs of installations and performance data, and with new sections on threading, use of this pipe in air conditioning, industrial maintenance and process uses, and in brewing and distilling applications.

No. 2020—POWER TRANSMISSION EQUIPMENT: Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. 208 p. catalog of power transmission equipment, featuring anti-friction roller bearing units, new group of babbitted bearings, new types of take-ups and other new products, as well as hangers, pulleys, gearing, couplings, etc., and giving dimensions, weights, list prices and other data.

No. 2021—PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 8 p. bulletin on propeller pumps for handling large quantities of water at low heads, showing features of construction sizes, capacities and representative installations.

No. 2022—PUMPS: Worthington Pump & Machinery Corp., Harrison, N. J. Series of 4 p. bulletins on (a) "Monobloc" self-priming pumping units, (b) "Freflo" centrifugal pumps for underpass drainage (c) multi-stage centrifugal pumps for boiler feed service up to 1000 lb per sq in., (d) vertical triplex single acting power pumps with V-belt drives, and (e) horizontal single piston pumps for general services of valve plate type, one type for pressures up to 75 lb and the other up to 250 lb.

No. 2023—REFRACTORIES: Standard Fuel Engineering Co., 667 Post Ave., S., Detroit, Mich. 52 p. handbook of information on refractories and refractory cements, with useful tabular material, photographs of typical installations, and instructions for applying high temperature cements by various methods.

No. 2024—SHEARS: Niagara Machine & Tool Works, 637-697 Northland Ave., Buffalo, N. Y. 26 p. bulletin on power squaring shears, capacities No. 10 gauge soft steel and lighter, with description of features and capacity tables.

No. 2025—STOKERS: Iron Fireman Mfg. Co., 3170 W. 106th St., Cleveland, O., 12 p. catalog describing pneumatic spreader stokers for bringing advantages of pneumatic conveying to steam size coal. Included are drawings of typical settings under various types of boilers.

No. 2026—STOKERS: Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. Two booklets on automatic firing with bituminous coal, one devoted to applications of this company's stoker to commercial heating plants, and the other to the automatic generation of high or low pressure process steam. Unit features an air meter which automatically meters the right quantity of air to the fuel bed at all times, and does not require manual adjustment in the field.

No. 2027—THREAD CUTTING EQUIPMENT: Landis Machine Co., Inc., Waynesboro, Pa. 16 p. booklet on performance of this company's thread cutting equipment, giving data obtained with threading machines, die heads and collapsible taps.

No. 2028—TUBING: Parker Appliance Co., 17325 Euclid

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INTERNATIONAL

Ave., Cleveland, O. 34 p. bulletin on seamless tubing, including stainless steel alloy, carbon steel, copper, high tensile bronze, monel metal and nickel tubing, dehydrated refrigeration tubing, oil burner tubing and other types, with data on selection and specification of tubing, thermal expansion, safe internal working pressures, and including list prices and other pertinent information.

No. 2029—UNIT HEATERS: Airtherm Mfg. Co., 1474 S. Vandeventer Ave., St. Louis, Mo. 16 p. bulletin describing and giving complete engineering data on "Skinner Brothers Type UA" floor type unit heaters, with capacity tables for various entering air temperatures and steam pressures, data on steam piping and automatic temperature control and dimension drawings and tables. The inside back cover is a heat loss calculation chart.

No. 2030—VALVES: Consolidated Ashcroft Hancock Co., Hancock Valve Div., 11 Elias St., Bridgeport, Conn. 20 p. bulletin describing new line of forged steel valves for steam pressures up to 1500 lb, including features intended to reduce valve replacement and maintenance costs. Available in globe and globe check types, with screwed and flanged ends. Construction is illustrated and described, and specifications, dimensions, list prices and weights are presented.

No. 2031—VALVES: Edward Valve & Mfg. Co., Inc., 1200 W. 145th St., East Chicago, Ind. 20 p. catalog on valve material specifications and standards, outlining service requirements for high temperatures and high pressures. Chemical analyses and physical rejection limits of recognized specifications are included, with annotations as to uses, limitations and variations from the basic standards. Alloying elements, corrosion resistance, heat treatment and dimensional standards are discussed.

No. 2032—VALVES: Fedders Mfg. Co., 57 Tonawanda St., Buffalo, N. Y. Issue of "Fedders News," Volume 3, No. 4, including an article by Joe Askin on performance characteristics of "Model 33" thermostatic expansion valves, devoted to effect of valves on the condensing unit.

No. 2033—VAPOR HEATING: Gorton Heating Corp., Cranford, N. J. 8 p. bulletin on this company's single pipe vapor heating system, describing radiator valves, equalizing valves and importance of equalization, and discussing valve area vs. free passageway.

No. 2034—VENTILATORS: The American Larson Ventilating Co., 1004 Keystone Bldg., Pittsburgh, Pa., 16 p. booklet on rotary suction type roof ventilators for ventilation of industrial and public structures, and for marine applications. Development and design of this company's ventilators is described, relative efficiency of ventilators is discussed, a typical solution for ventilating problems is worked out in detail and specifications for ventilators, bases and dampers are given. In

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1008	1009	1010	1011	1012	1013	1014	
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2003	2004	2005	2006	2007	2008	2009	2010
2011	2012	2013	2014	2015	2016	2017	2018
2019	2020	2021	2022a	2022b	2022c	2022d	2022e
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2031	2032	2033	2034	2035	2036	2037a	2037b
2037c	2037d						

Name ..... Title .....

Company .....

Address .....

City ..... State .....

technical manner the scientific development of this company's formula for ventilator design is presented and supplementary engineering data are included.

No. 2035—WATER LEVEL ALARMS: Reliance Gauge Column Co., 5902 Carnegie Ave., Cleveland, O. 8 p. catalog on float operated safety water columns for boilers, and technical specification and installation data on patented monel metal floats. Selection of proper alarm columns to meet definite requirements is outlined.

No. 2036—WELDING FITTINGS: Tube-Turns, Inc., 425 S. Fifth St., Louisville, Ky. 48 p. catalog and list price containing data on "Tube-Turns," tees, welding neck flanges, Van Stone flanges, slip-on flanges, threaded companion flanges, blind flanges, heads, reducers and shaped nipples. Advantages of welding fittings and welding are discussed, data are given on construction features and frictional resistance, and tables on pressure loss through elbows and loss by pipe threading are shown. Pressure-temperature ratings of these fittings are tabulated, and installation photographs are shown.

No. 2037—WROUGHT IRON: A. M. Byers Co., Clark Bldg., Pittsburgh, Pa. (a) 16 p. booklet on wrought iron in industry, describing the requirements to be met, resistance of this material to corrosion and its other features, and showing installation views of wrought iron for meeting industrial requirements; (b) 16 p. booklet on the use of wrought iron in salt water services, salt water being one of the most active natural corroding mediums to which metals must be subjected, discussing the application of wrought iron to this service, presenting results of experience and showing numerous uses; (c) 16 p. booklet on the use of wrought iron in refrigeration systems, presenting information on the corrosive conditions to which various parts of refrigeration systems are subjected and giving service records on wrought iron; (d) 16 p. bulletin on the use of wrought iron for flue gas conductors and coal handling equipment, outlining the corrosive conditions met with in this service and including a special article on smoke stacks giving suggestions covering specifications and design details for stacks.

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